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COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 5, No. 11 MAY, 1934 25c a copy

MAY 1 1934

Connors Creek Station, Detroit Edison Company

Heat Cycles—A Symposium

Plant Engineering—A Message to Engineering Students

MODERN STEAM PLANT EQUIPMENT

No. 3 *of a series presenting design and operating features of C-E products*

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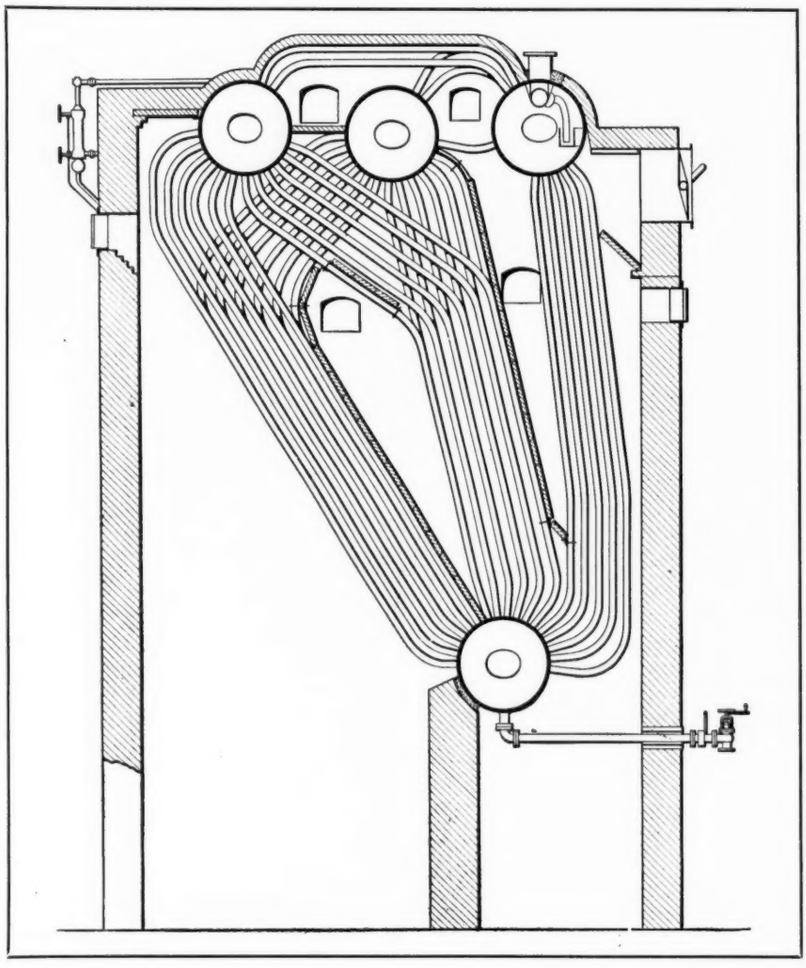

FEATURES

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Steam Quality . . . With proper condition of feedwater, the design of this boiler limits the moisture content in the steam to a small fraction of 1 per cent.

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COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME FIVE NUMBER ELEVEN

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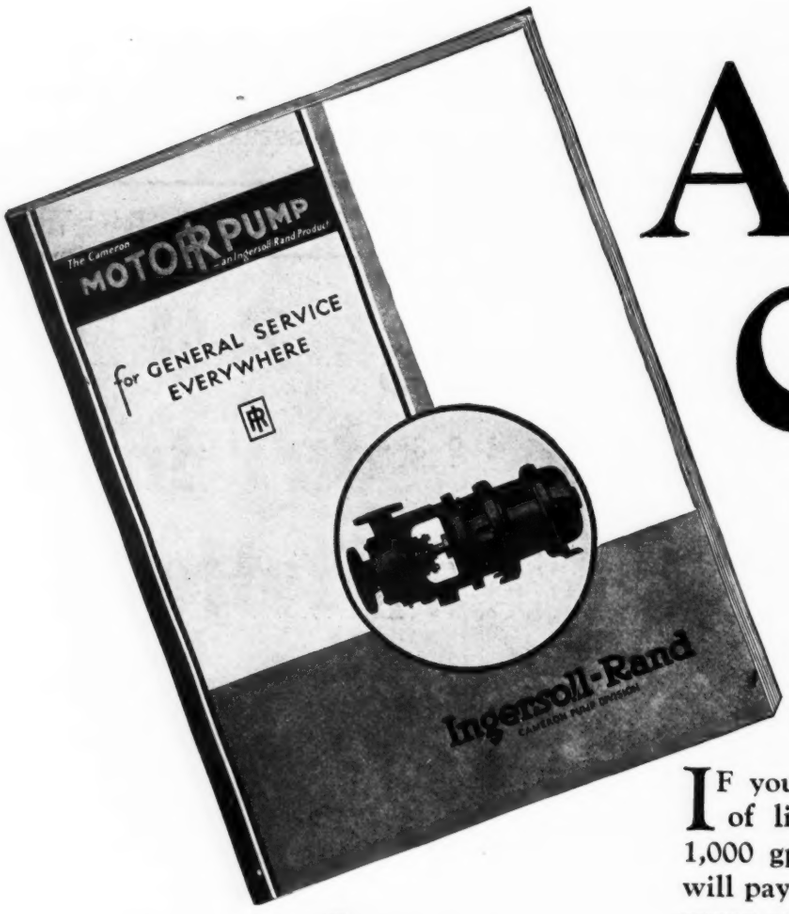

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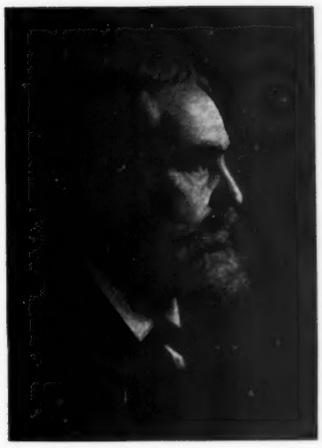


OSKAR VON MILLER

(1855-1934)

Exzellenz Dr. ver. pol. h.c., Dr. Ing. e.h. Geheimer Baurat, Zivil Ing., Ing. Büro, Director of The Deutsches Museum of Munich

DR. VON MILLER, had he lived another month, would have completed seventy-nine years of one of the most useful lives of this "century of progress." He had attained world-wide fame before the undertaking of the Museum by which work he is best known. He was the moving genius of the project of the Frankfurt-Lauffen transmission of power at 25,000 volts in connection with the electrical exhibition in Frankfurt in 1891, of which he was president. Previous to that he had visited the United States, met Edison, returned to Germany and in 1883 was a director of the German Edison Company that installed in Berlin the first municipal electric lighting system in Germany.

It was in 1878 that Dr. von Miller received, on a visit to London, his first impulse to erect in Munich a museum similar to the Science Museum. However, it was not until 1903 that he left his commercial engineering work as a major activity and, at an age approaching fifty, undertook to devote himself to the Museum which is the most remarkable of its kind in the world. Although previous to that time his professional work had been in Berlin, actuated by the ambition to make his native city, Munich, the center of knowledge and art, he left Berlin to institute the Museum.

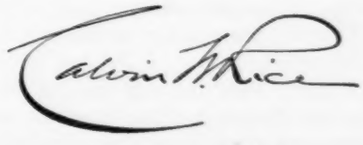
One of the features of the Museum is that all exhibits, so far as possible, are working models to be actuated by the observer. Instead, therefore, of signs "Do not touch," every one is told how to operate the mechanism. It is, therefore, a children's paradise. School children from all over Bavaria, and other parts of Germany, visit the Museum. As a consequence, in a city of less than one million inhabitants more people visit the Deutsches Museum annually than any other museum of any kind, in the world. One of Dr. von Miller's principles was to make a small charge, corresponding to about five cents in our money, as a result of which the museum is nearly self-supporting.

When there is a life of such power and influence it is a matter of more than interest to study how it originated and was developed. He was the tenth child in a family of fourteen. His father was an artist at the head of a bronze foundry. The fountain in Fountain Square, Cincinnati, was made by the father, Ferdinand v. Miller and installed by the son Oskar v. Miller. The mother came from an especially able family of high social and governmental standing. In the will of Ferdinand, the father of Oskar, was the admonition for each member of the family to love and assist the others throughout life.

Oskar as a young lad was pushed out to support himself. When he was married it was in a little church in Neuhausen. Every year since, that is, for over fifty anniversaries, the entire family has held a religious service in the selfsame church. Every two weeks the family has always met for a day or an evening together.

Combined with Oskar v. Miller's great attainments was his love for celebrations. On May 7, his birthday, all functions of the Deutsches Museum were celebrated and the whole city of Munich on one occasion made it a holiday. Exzellenz v. Miller regarded his life as belonging to the people and repeatedly excused himself from his family to receive people, stranger or otherwise.

The Museums of Science and Industry in New York, Chicago, Philadelphia, Detroit and the projected museums in other cities, all were inspired by the Museum in Munich.



EDITORIAL

Stimulating Engineering Society Membership

Practically all the engineering societies, national and local, have suffered during the past three years. Memberships have fallen off and wide unemployment within the profession has made the collection of dues difficult. Yet, despite these handicaps, and the added burden of unemployment relief, the technical work of the societies has, in general, gone forward in a most creditable manner with but slight curtailment.

Now, with indications of improvement within the profession, it is opportune to re-enlist the interest of those who have ceased active participation or even let their soceity memberships go by default. In this connection, the present efforts of the Executive Committee of the Metropolitan (New York) Section of the A.S.M.E. deserve attention.

Recognizing the great diversity of interests among a group of nearly five thousand, the Committee set about to canvass these members, both active and delinquent, as to their professional interests, their hobbies, social desires and views as to how the Society could best meet their needs. The responses, numbering approximately nine hundred, have been most illuminating and clearly indicate the need for breaking up the activities into small specialized groups. Moreover, a consensus of the replies stress the value of acquaintanceship and social relations as an adjunct to technical activities.

Proceeding along the lines of these suggestions the Committee has inaugurated a program which, thus far, is meeting with gratifying results. Many whose active services and association were in danger of being lost to the Society have renewed their interest and are enthusiastically cooperating in carrying out the program. As the work proceeds it is anticipated that a still larger number will respond.

Engineering groups in other sections of the country and in other Societies might well profit by this experience as an aid to rehabilitating membership and reviving interest in society activities. The Committee has issued a bulletin which summarizes the results to date.

Plant Engineering

The season is at hand when 1934 engineering graduates must look to securing positions. As in the past two or three years, some will undoubtedly be forced by expediency to accept work in lines other than those for which they have been trained. In some cases this diversion of purpose will be temporary while in others it is likely, by force of circumstances, to alter the graduate's life work.

With men thus drifting out of the profession and graduating classes smaller, there is basis for the belief that shortage of junior engineers will exist when industry has again got into full swing.

The curricula of engineering schools have, within recent years, stressed the breadth of technical education, but the practical education should be as broad as the theoretical. It is surprising that more schools have not seen fit to include courses in plant engineering, a field which offers a wide diversity in practical experience. It includes power generation and application, mechanical equipment of all kinds, some design and construction, cost accounting, occasional chemistry and a multitude of problems peculiar to the particular industry concerned. This is well brought out by the discussion of Mr. Gibson in this issue.

It may be contended that most of these subjects are dealt with individually at some time or another during the university course, but apparently little attempt is made to correlate them into the duties of plant engineering, such as would popularize it in the mind of the student. Yet many engineers have risen to positions of importance and broad responsibility through this line of endeavor.

With production picking up in numerous industries, accompanied by a certain amount of rehabilitation of plant facilities, there should be opportunities for placements in the plant engineering departments of industrial establishments. In view of this the young engineer would do well to give serious thought to this field.

The Federal Power Survey

About eight months ago legislation was enacted authorizing the Federal Power Commission to undertake a nation-wide power survey. Until quite recently little was heard of the progress of this survey, but it now appears that in the interim the Commission has been getting together the necessary organization, inventorying the available information existent in Federal and State agencies, and establishing contacts with public and private bodies that are working along these lines.

This procedure is commendable and will save much needless duplication of effort, such as would have resulted had the Commission plunged into the field work of the survey immediately.

The two major objectives of the investigation will be: First, to ascertain, if possible, the present and future power requirements of the United States; and, second, to determine from what sources these requirements can be supplied most economically. In addition it will compile the rates charged for electricity by both municipal and private utilities.

The results of this survey should prove useful in long-time planning. It is unfortunate that they were not available a few months back when they might have exercised a restraining influence on Public Works expenditures for certain hydro projects the economics of which at this time are questionable.

Heat Cycles and Their Relative Efficiencies

A symposium of opinions by A. G. Christie, A. D. Bailey, Frank S. Clark G. A. Hendrickson and Alfred Iddles, discussing the article by F. H. Rosencrants under this title in the April issue of Combustion.

Discussion by A. G. CHRISTIE

Professor of Mechanical Engineering
The Johns Hopkins University

THERE are few subjects of more absorbing interest to power engineers than discussions of heat cycles and their relative efficiencies. Frequently these discussions become quite mathematical and hence do not appeal strongly to the busy operator. The papers[1] of Mr. Rosencrants which deal with heat cycles, have been written in a clear and lucid style and can be readily followed. His conclusions, based upon his studies of the various cycles, have considerable influence on American steam station development.

Mr. Rosencrants' contribution in the April issue of COMBUSTION presents comparisons between three present day cycles and a high temperature cycle. His results favor his new "Cycle *P*" which is for 800 lb per sq in. absolute, 925 F, and 0.5 lb absolute back pressure with four stages of regenerative feedwater heating but no inter-stage reheating. A heat consumption of 11,400 Btu per kilowatt-hour output is predicted for this station. With 13,700 Btu per pound this represents a coal rate of 0.83 lb per killowatt-hour output which is a low figure. Many engineers will recall that about twenty-five years ago, the gas producer and engine were widely heralded as the last word in efficiency with a heat consumption of "a pound of coal per horsepower-hour." How inefficient such a figure now seems! As one looks back over the past twenty years and notes the truly phenomenal improvement in the thermal performance of steam stations, one hesitates to make predictions of future progress or of ultimate performances. It is quite possible that still further gains may be achieved by means of cycles and plant constructions as yet unconsidered.

Mr. Rosencrants adopts a conservative attitude toward future cycles. His cycle *P* is quite within the range of possibility and involves only a comparatively small temperature increase over cycles now in use. The simplicity of this cycle for straight condensing service will strongly appeal to operators. Its construction will be less expensive than high-pressure reheating or bi-fluid plants. In view of the success attained with the 1000 F plant by the Detroit Edison Company there should be little hesitation in using the more moderate temperature of 925 F.

Reliability also requires consideration. There is nothing to fear from the pressure of 800 lb since stations operating with 1200 lb have developed no troubles from pressure alone. While the temperature of 925 F is above that commonly used in present-day plants, materials are available to withstand successfully this temperature. Prices of such materials will fall with increased use of these new alloys. Certain turbine builders are prepared to furnish units for this temperature and reliable operation may be expected. One must remember that with throttling governing, the temperatures of the high-pressure stages vary only a small amount with wide changes of load. This lessens the risk of accident from temperature variations resulting from load changes.

Engineers should not overlook the last paragraph of Mr. Rosencrants' article. It appears that electrical utilities will be subject for some time in the future to high taxation, public ownership competition and drives for lower rates. Hence it will be imperative to search out every means that will reduce power costs. In many cases the addition of superimposed 800-lb plants on the older 250-lb stations will provide plant extensions at lower costs than otherwise when full consideration is given to plant capacity factors over the whole useful life of both pressure equipments. Many such additions may be expected as load growth is resumed.

Let us consider cycles that may provide still better performance than cycle *P*. Based on the experiences of the Detroit Edison Company with 1000 F steam temperature, this may be considered as a possibility within the near future. With this temperature the inlet pressure may be raised to 1200 lb per sq in. abs which with five stages of bleeding and no interstage reheating, indicates a possible performance of about 10,900 Btu per kilowatt-hour using the same efficiency assumptions as in Table 1 of Mr. Rosencrants' paper.

There are boiler plants in America which are now

[1] See COMBUSTION, May 1931 and April 1934.

operating at higher annual efficiencies than 87 per cent, as assumed by Mr. Rosencrants in Table 1. Published data on tests of the new Velox boiler recently developed by Brown Boveri & Company of Baden, Switzerland, show efficiencies as high as 91 per cent. There are indications that higher values may be attained. If one can assume a boiler efficiency of 90 per cent, the cycle performance of the 1200-lb, 1000 F plant without reheating, becomes 10,550 Btu per kilowatt-hour or about 0.76 lb per kilowatt-hour with coal of 13,700 Btu per lb. Such a plant possesses the simplicity of the 800-lb plant with 7 per cent better efficiency. The plant with 1200 lb, 1000 F is also about 7 per cent more efficient than the 1200 lb, 800 F plant with 800 F reheat.

A few years ago much attention was devoted to single-flow types of boilers for very high pressures and the performances of some experimental units have been reported. Benson boilers of this type for 3200 lb pressure are at present in continuous operation in several foreign plants.

A recent development in the use of this particular boiler demands the attention of power-station engineers. Mr. G. Gleichman of Siemens Schuckert of Berlin, Germany[1], reports the successful operation of a Benson boiler and its turbine under novel conditions. Instead of throttling the steam to the turbine at low loads, the boiler pressure itself was lowered. In other words, the set operated with constant throttle temperature and variable boiler pressure. The condenser would be designed for a comparatively low boiler pressure and would be overloaded at the higher pressures. This leads to nearly constant volume at the turbine throttle and the weight of steam admitted is nearly proportional to the boiler pressure. There appears to be about the same Rankine heat per pound available at all pressures, assuming a loss of vacuum at the higher loads. Hence power output is proportional to steam flow and, in turn, to rate of fuel firing. The final governing will be the control of fuel feed. The turbine efficiency is maintained over a considerable range of load under these conditions.

Claims are made that the Benson boiler can be built for less money than any other type. This system appears to have interesting possibilities not only as an efficient cycle but as a low cost plant. Incidentally, it is said that the rights to build this system in America have recently been secured by one of the turbine builders.

The load decreases of the past few years and the public reaction against privately owned electrical utilities, have caused some engineers to view the future of the industry with pessimism. They cannot conceive that larger and more efficient stations must be built in the near future. The depression will not last forever and the lower rates of the future will induce a greater consumption of electricity than ever before. If our average user only approaches the use of electricity of the Canadian consumer our plants must be extended by at least fifty per cent. On the other hand, it has been past experience that the increase of electrical consumption is very rapid following a period of stagnation. We may therefore look forward to renewed activity in the construction of new power plants and to the early application of the new cycle proposed by Mr. Rosencrants, and even of higher temperature cycles.

[1] Excerpts from Mr. Gleichman's paper will appear in the June issue of COMBUSTION—*Editor*.

Discussion by FRANK S. CLARK

Consulting Engineer
Stone & Webster Engineering Corporation

THE author, both in his article in COMBUSTION for April 1934 and in the preceding article in May 1931, has set forth clearly and concisely the relative theoretical advantages of the various accepted and projected power producing cycles. He prefaces his 1931 article, however, by stating that "any discussion of initial cost, of maintenance expense and of availability of equipment is necessarily omitted." These items, particularly the first one, loom large in the eyes of utility men and must be considered in any application to be made.

Any radical departure from normal practice must expect to experience a certain amount of "birth pains" in the nature of increased construction and maintenance costs and operation difficulties. These may be expected to decrease with experience and must be balanced against the possible efficiencies discussed by Mr. Rosencrants. The industry is indebted to such companies as the Edison Electric Illuminating Company of Boston, the Detroit Edison Company and the Hartford Electric Light Company who have pioneered in fields of high pressure, high temperature and the binary-vapor cycle.

For any given capacity and for any conditions now usually considered, installation costs increase with greater efficiencies. These increases are uniform and proportionately small so long as the same operating cycle is adhered to and as long as no special materials or construction are required. When these are necessary, and when the operating cycle becomes more complicated, the installation costs jump disproportionately.

The regenerative cycle has been in use about thirteen years, and is standard and accepted practice in modern power stations. Its limitation is the steam temperature which can be used safely with the metals that are available. We have now arrived at a point where 850 F is being used without employing expensive alloy metals. The corresponding efficient pressure is about 650 lb. Stations now operating under these conditions have shown efficiencies comparable with those given for the high-pressure reheat cycle. These efficiencies at 850 F are attained at but moderate increment investment costs over those for the 450 lb, 750 F regenerative cycle. At present, an increase in temperature above 850 F involves the use of more expensive materials whose economic application has not been proved. I believe with Mr. Rosencrants that 925 F, with its accompanying pressure of 800 lb, is probably the next logical point in the development of the regenerative cycle, but we should expect to encounter greater difficulties as we increase temperatures.

The reheat cycle using 850 F temperature and 1400-lb pressure has been adopted for the new Port Washington Station and for the new units at State Line. Mr. Rosencrants states it is probable that few, if any, more plants of this type will be built. I believe, however, this cycle and pressure to be worthy of careful consideration in the case of the rehabilitation of existing low-pressure plants where higher pressure results in increasing the capacity of the superimposed equipment, as well as in greater efficiency for the whole plant.

The binary-mercury vapor-steam cycle offers oppor-

tunity for efficiencies exceeding the straight steam cycles and the results at Hartford have fulfilled expectations. After a series of birth pains, the installation has been performing for over a year with a record for reliability equally as good as any steam power station and at a net Btu per kilowatt-hour about 10,500. The installations made so far have been in connection with existing steam-electric generating stations or for the purpose of generating steam for industrial uses, and, to the writer's knowledge, no accurate judgment can be formed as to installation cost when made in connection with a new station and solely for power producing purposes.

Discussion by G. A. HENDRICKSON
The Detroit Edison Company

IN STRIVING toward a "best" heat utilization cycle a power-plant designer must bear in mind two principal elements of the problem; first, the thermal conditions required for the attainment of a high overall efficiency, and second, the cost of producing those conditions. A proper economic balance between these two factors is the ultimate aim of every design. Naturally, as manufacturing methods improve and new materials become available to the power-plant designer at new prices, the point of most favorable balance changes and plant construction must change accordingly. These adjustments involve the whole field of engineering: the metallurgist is asked to develop alloy metals to withstand higher pressures and temperatures, shop methods must be improved to make possible the fabrication of these new alloys at reasonable prices, etc.

In the past such adjustments have come in an orderly step-by-step progression with ample time to prove each development before others of importance were proposed. With this in mind it is readily appreciated that the situation at present is more complex than usual. Before a definite practice could be established in regard to the reheat cycle serious consideration was given to the adoption of an alternative method for obtaining the same efficiency, namely, an increase in initial temperature. This situation puts two quite different types of plant in direct competition for the favor of power-plant operators.

An unbiased view of the situation leads to the conclusion that each of these plants has its own points of merit and consequently each has its own field of application. Both should, therefore, be examined closely to determine their fitness for present needs and their possibilities for future development. A few words on the characteristics of a "best" heat utilization cycle may facilitate such an examination.

It is fundamental that, for high efficiency, heat be supplied to a heat engine at the highest possible temperature; for the supply temperature determines the proportion that must be rejected as waste heat. This is a simple though incomplete statement of Carnot's principle, and it is equivalent to saying that, in a steam power plant, the bulk of the heat should be supplied to the steam at the highest possible temperature. Since the supply temperature determines the proportion of heat rejected, the benefit of increased temperature is readily seen. There is a pressure effect, however, that is not so apparent, because it works through temperature. An increase in temperature by adding superheat, for example, increases the temperature at which only the last portion of heat is added. The greater part, namely, the latent heat of vaporization, is still added at the same temperature as before, that is, the vaporization temperature corresponding to the pressure in use. This temperature can be raised only by raising the initial pressure. Thus a heat engine using vapor media requires both high initial temperature and high initial pressure for the highest efficiency.

Efforts to obtain these conditions have led to the two types of cycle mentioned, high pressure and moderate temperature with reheat, or moderately high pressure and a somewhat increased temperature without reheat. Both of these schemes aim at the same result, i.e., an increase in varying degree of the vaporization temperature or heat supply temperature. Since an increase in vaporization temperature means an increase in initial pressure, it means also an increase in moisture content in the exhaust. There is a practical limit, however, to the concentration of moisture that can be passed through the blading of a turbine without excessive erosion. Most modern plants operate close to this limit which may be placed in the vicinity of 12 per cent. A further increase in initial pressure, therefore, requires the introduction of methods for reducing the moisture content of the turbine exhaust. In both cases cited the reduction in moisture is accomplished by adding more heat per pound of steam. In the reheat cycle it is added as reheat and in the non-reheat cycle as initial superheat.

The thermal aspect of present fitness is readily determined from the results of a paper published in the April 1934 Transactions of the A.S.M.E.[1] From this study it appears that at a given pressure one stage of boiler-room reheat produces the same increase in efficiency as a 125 or 150 F increase in initial temperature. Thus, to fit present needs most satisfactorily, the question must be answered whether it is more economical on the one hand to build three steam lines between boiler room and turbine room or on the other to install alloy steel in those parts where the temperature much exceeds 850 F. This would include the last part of superheater, the steam main and the turbine bowl. Since the superheater surface is about the same in either case the choice lies between specifying alloy steel for the parts affected by high temperature, or installing two extra steam mains to avoid those high temperatures. Either choice is attended with its own peculiar difficulties and cost items.

At present, in plants with average use factor, the writer believes that the economic balance is in favor of higher temperature without reheat. Also, for the future, since a higher temperature increases the margin between plant efficiency and the Carnot limit, this cycle offers the greater opportunity for further development and is therefore more desirable. Some time may elapse, however, before the full benefits of increased temperature are commercially attainable. In the meantime, the great reduction in heat rate immediately available to the reheat cycle at temperatures suitable for present day commercial materials, makes reheating attractive for base-load plants where a high use factor affords good opportunity to pay out on the greater investment.

As regards the steam conditions available to higher temperature cycle it is probable that 850 to 900 F is the

[1] A Thermal Study of Available Steam Power Plant Heat Cycles by G. A. Hendrickson and S. T. Vesselowsky. A.S.M.E. Transactions, Vol. 56, pages 203–224, April 1934.

maximum temperature that can now be handled economically. At this temperature the pressure limit imposed by the moisture content of the turbine exhaust is about 600 to 800 lb per sq in. at the turbine throttle. Thus 900 F and 800 lb is suggested as the best cycle at present for non-base-load steam power plants.

Discussion by A. D. BAILEY

Superintendent of Generating Stations
Commonwealth Edison Company

ALTHOUGH few large stations have been built during the last few years, it is nevertheless interesting and necessary to contemplate what we would do in the way of design if, as, and when the opportunity to build a new station should arise. If any one feature of station design can be predicted with certainty, we can be sure that simplification will be given the greatest consideration. The reason is obvious as this insures minimum investment cost and minimum operating cost, at least so far as labor and maintenance are concerned. With this in mind, a steam cycle which carries out the simplification idea and which gives very high efficiency is sure to receive quite favorable consideration. Mr. Rosencrants' prediction is a look ahead.

While high steam temperatures offer the greatest hope for improved station economy, the limitations imposed by the materials of construction have made it necessary in the past to adopt the reheat cycle in order to reduce operating costs. Experience with steam pressures in the 1200 to 1400-lb range has shown that pressure itself is no problem as the operating records of these installations have been favorable. There has been little experience, however, with 925 F steam installations, probably none in large units. So long as manufacturers recommend the added complication of superheat control in order to prevent over-stressing of metals, station designers and operators are going to be slow to accept the higher temperature. Unfortunately, there are only a few stations in operation using temperatures in excess of 800 F, and it seems to be the impression that any temperature above this is hazardous unless the machine parts subjected to this temperature are small and constructed of very special and expensive materials.

It must not be forgotten that each installation presents peculiar conditions and problems so that general conclusions cannot be satisfactorily predicted. For economical reasons there may be many superimposed installations installed in existing moderate pressure plants, and with simplicity and consequent economy in mind, an effort will probably be made to get the greatest amount of power and efficiency possible with the already existing equipment, or with as few changes as possible.

The improvements which have been made in steam stations in the last generation are a good background for favorable predictions for the future. With intensified research, materials will doubtless be developed to meet any demand which industry may make, particularly when the amount of money involved justifies a development. Equipment for temperatures in the 900 F range is being offered by manufacturers, and with the resumption of station construction, it is certain that the economical possibilities of this temperature will demand careful consideration.

Discussion by ALFRED IDDLES

Vice President, United Engineers & Constructors, Inc.

The articles by F. H. Rosencrants, published in COMBUSTION, May 1931 and April 1934, constitute a comprehensive review of the present status of power-station heat cycles as they stand today. The characteristics and broad economic factors of these heat cycles have in the main been established in practice. The binary cycle, of which mercury-steam is the most developed instance, must still be considered in the development stage. Difficulties are still being met with of sufficient moment that it is impossible to say that this type of cycle is completely satisfactory.

Probably the most generally accepted heat cycle today is the simple high-pressure regenerative arrangement without reheat. It is still questionable whether in the final analysis either steam or gas reheat constitutes an economically profitable development as compared with the high-pressure regenerative cycle. While it is true that, at least with materials at present available, the simple regenerative cycle cannot be carried beyond a temperature of 850 to 900 F, which necessitates the limitation of pressure to 700 or 800 lb. on account of moisture content in the lower stages of the turbine, the gain in efficiency secured by going to the higher pressures with reheat is nearly if not entirely offset by the extra cost of reheating and related equipment.

The problem confronting designing engineers today is more often one of using a cycle arrangement for installation in connection with a previously existing power system. A large proportion of the total generating capacity in operation today is working with steam conditions which, as Mr. Rosencrants has pointed out, were considered good practice up to 1923 or thereabouts. At that time the art had not developed to a point where conservative engineers were inclined to adopt temperatures much in excess of 700 F or pressures above about 400 to 450 lb. Today there is little hesitancy in using temperatures up to 900 F and pressures up to 850 lb. Turbine manufacturers have indicated their willingness to build for temperatures of 1000 F with 1000 lb pressure. With such steam conditions the simple regenerative cycle has efficiencies comparable with those of the binary cycle and apparently the construction costs will be somewhat less than for the binary cycle. Certainly assuming that materials are available which will satisfactorily hold these pressures and temperatures, the operating complications will be less.

This situation brings out into bold relief the relatively poor efficiencies of the majority of the power stations operating today. Practically every operating company is giving consideration to means for improving the efficiency of existing plants in connection with their increases in capacity. This type of installation has been made in a number of cases and in all probability will be carried further, particularly since pressure and temperature limits have materially increased.

It is, of course, impossible to lay down any general rules for the improvement of an existing station by imposing upon it a high-pressure element or elements, which will exhaust into and supply steam for existing prime mover equipment. The particular cycle used must be tailor-made to fit the peculiarities of each case.

Air Leakage in Steam Generating Units

By W. S. PATTERSON, Combustion Engineering Company, Inc.

PART III

ITS MAGNITUDE AND ECONOMIC IMPORTANCE

In Part I of this series[1] it was stated that "the seeming unbalance of heat input to heat output sometimes found in the air heater may be used as a means for calculating the percentage of air infiltration into the furnace and boiler setting." In Part II[2] it was shown why the heat absorbed by the air heater as calculated by the A.S.M.E. Test Code method will not balance, except by accident, with the difference between the heat content of the flue gases entering and leaving the heater. In this, the final article of the series, equations are developed to predict the change of leakage in the parts of a steam-generating unit as the evaporation rate is varied. The results of tests are presented and discussed.

BEFORE proceeding further with the subject of air leakage let it be noted that all the views expressed in this series by the author, and generally accepted by his associates in this country, are nevertheless not shared by some others, notably in Great Britain. A technical treatise recently published[3] in London attributes the seeming unbalance between the heat lost from the gas and heat absorbed by the air to the hypothesis that solid carbon burns within the air heater, liberating heat which is not accounted for in a mere balance of the sensible heat of the fluids entering and leaving the apparatus. Exponents of this theory have found, as we have, that the balance of heat absorbed is closer for stoker-fired installations than for pulverized-fuel units. They explain this on the basis of less solid carbon being carried through the heater in the flue gases of stoker-fired units. This hypothesis is discredited by the author insofar as the magnitude of its effect on the air-heater heat balance is concerned. It will not be considered further in this discussion.

Variation of Leakage with Change in Load

Infiltration into the furnace is directly proportional to the product of the leakage area and the square root of the differential pressure, the general equation for flow through orifices being:

$$Q = 1096.5\ CA\ \sqrt{\frac{P}{d}} \qquad (1)$$

in which Q represents the discharge in cubic feet per minute when P is the differential pressure head in inches of water, d the density in pounds per cubic foot, A the leakage area in square feet (which in the case of a furnace consists of the total area of all the cracks and other sources of leakage), and C the average discharge coefficient for the leakage orifices or cracks.

In a given installation consider the variation in Q with evaporation rate of the boiler. Obviously, neither the average temperature of the air outside of the furnace nor the nature of the leakage cracks will vary much. Therefore, C and d may be considered as constant. Offhand, one would say that A would also be constant but this is not necessarily true. If a large portion of the area consists of expansion cracks in brickwork, which enlarge in area at reduced load as the walls cool, the value of A may actually increase slightly. The pressure differential across the furnace wall, P, varies from the top to the bottom of the furnace due to the stack effect of the hot gases. In equation (1) it would be the mean pressure differential for the infiltration area A and would change as the temperature of the gases varies with the evaporation rate. However, the decrease in static draft for a change in gas temperature from 2500 F to 1500 F would be only about 10 per cent. Furthermore, it was pointed out in a previous article[1] that at reduced load there is a tendency, with manual operation, for the furnace draft to be actually higher than at full load. It is therefore logical to assume that the quantity of furnace infiltration at low rates of evaporation at least will equal the quantity at high rates of evaporation, since it has been shown that under certain conditions it might be slightly greater.

The total combustion air supplied to a steam generating unit, whether measured by gas analysis in the furnace or at the boiler outlet, is directly proportional to the product of the fuel combustion rate and the excess air coefficient. Therefore, as the fuel burning rate is decreased more and more, the constant quantity of furnace infiltration becomes a larger and larger percentage of the total air supply. It is most significant to visualize furnace infiltration as a percentage of the total air supply because by so doing its immense importance at low combustion rates can be understood best. It may be plotted graphically against combustion rate of fuel in

[1] Air Leakage in Steam Generating Units, COMBUSTION, February 1934.
[2] Air Leakage in Steam Generating Units, COMBUSTION, April 1934.
[3] "The Thermal Balance of Air Preheaters of Large Steam Generators," Dr. M. W. Travers, F.R.S., "Engineering" (London), Aug. 5, 1932.

pounds per hour, pounds per square foot of grate, pounds per cubic foot of furnace volume or Btu per cubic foot of furnace volume as in Fig. 1 provided the excess air coefficient correction is properly made. Fig. 1 shows this relation for various quantities of furnace infiltration.

Consider next the air infiltration into the boiler setting. It is caused by the pressure differential across the setting walls which occurs principally because of the resistance offered by the boiler heating surface and baffles to the flow of the gases. As in the case of the furnace infiltration, the quantity will depend on factors given in equation (1) which may be modified, in accordance with previous discussion, to the form:

$$W = KA\sqrt{P} \qquad (2)$$

in which W is the infiltration in pounds per hour; A, the leakage area; P, the mean pressure differential; and K, a constant combining the constant of equation (1) with the mean values of C and d which may be assumed constant for a given installation. But since the mean pressure differential, P, in equation (2) is directly proportional to the boiler draft loss, it must also be proportional to those factors which affect the boiler draft loss, and for a given boiler we may therefore write:

$$P = K_1 \times \left(\frac{M^2}{d}\right) \qquad (3)$$

in which P is the mean pressure differential across the boiler setting walls; M, the average mass flow of gas through the boiler; d, the average density of boiler gases; and K_1, a constant combining the constant for the boiler draft loss equation with a proportionality factor for P, since P might, for instance, be equal in some cases to one-half the boiler draft loss.

It can be shown that for a variation in boiler evaporation rate of 5 to 1 the error in using an average value for d after combining equations (2) and (3) will amount to about 1 per cent of the total air weight supplied up to the boiler outlet. This is negligible considering the accuracy with which infiltration can be measured, so combining equations (2) and (3) and throwing d into a new constant, K_2, we have:

$$W = K_2(AM) \qquad (4)$$

For a given installation, with A practically constant at all rates of evaporation, the boiler setting infiltration, therefore, will be proportional directly to the weight of the gases passing through the boiler. Consequently, it will be a constant percentage of the total air weight regardless of the variation in excess air coefficient for a given fuel or the variation in rate of evaporation.

Factors Affecting Variation of Leakage Beyond the Boiler

Beyond the boiler outlet the variation of air leakage into the system depends largely on the method used for regulation of the furnace draft. It also depends somewhat on the characteristics of the induced-draft fan and on how closely predicted conditions of operation (i.e. volume, pressure and temperature) are approached by the actual operating conditions. Under ideal conditions it is possible that a properly chosen fan may be regulated, by variable speed control, so that its pressure-volume characteristic coincides almost exactly with the characteristic of the steam-generating unit. Under such

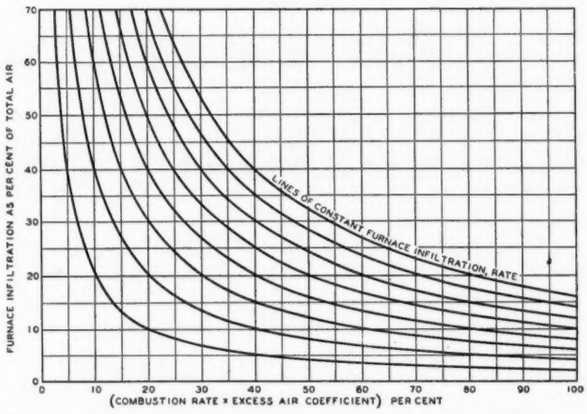


Fig. 1—Curves showing variation of furnace infiltration, per cent, for constant furnace infiltration rate, pounds per hour, as affected by changing rate of operation of furnace

conditions the negative pressure existing in the system would not be greater at any point than the pressure required to cause the flow of gases with all dampers wide open. For example, the mean differential pressure across an economizer casing, would be a certain fraction of the total draft loss of the boiler plus economizer and would vary in accordance with equation (3). This would make the infiltration a variable quantity but a constant percentage of the total air supply as indicated by equation (4).

When the induced-draft fan is operated at constant speed and the flow of gas through the system is regulated by operating a damper, the percentage of infiltration beyond the boiler may actually increase as the rate of evaporation is decreased. This condition would occur when the control damper is located at the boiler outlet, but under such circumstances the variation of leakage into the air heater or the economizer cannot be covered by a general equation. The pressure characteristic of the fan is the governing factor; the steeper the pressure curve the greater the increase in rate of infiltration with decreased rate of evaporation. If draft control must be accomplished by dampering it should be done at the fan inlet or outlet, not at the boiler outlet, except, of course, when there is no economizer or air heater and the fan is located close to the boiler.

Variation of Tempering Air for Pulverizers

The foregoing discussed sources of leakage will be found to a greater or lesser degree in every steam-generating unit no matter what type or what method of firing is used. The item of "tempering air" supplied to the pulverizer or feeder blower is peculiar to the pulverized-fuel fired unit. Tempering air is used when there is an air heater and when air from it is too hot to be used directly at the pulverizer or feeder, due to fire hazard. This depends on the kind of coal and design of the pulverizer or feeder. With certain fuels and certain types of pulverizers trouble may be encountered when the coal-air mixture leaving exceeds 125 F; with others 350 F involves no hazard.

The type of pulverizer also largely governs the quantity of primary air required to sweep the fine coal from the pulverizer and to carry it to the burner. Obviously, the lower the primary air quantity required to sweep the mill and the higher the allowable air temperature

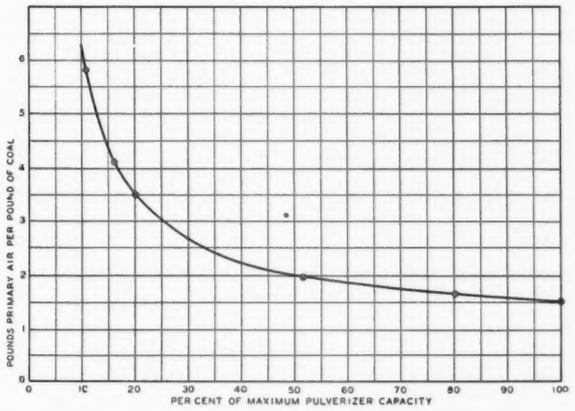


Fig. 2—Primary air characteristic curve for a type of pulverizer commonly used for direct firing of pulverized coal

entering the mill, the lower will be the quantity of tempering air required.

Tempering air robs the air heater and by so doing greatly decreases its heat recovery efficiency. Fig. 2 shows a typical primary air curve for a type of pulverizer commonly used for direct firing. It illustrates how the air requirements, per pound of coal handled, increase as the load decreases. At 10 per cent of maximum capacity the primary air supply would be about 50 per cent of the total air supply for an average coal, assuming it were possible to operate with low excess air in the furnace. But referring to Fig. 1, if the furnace infiltration were only 5 per cent of the total air supply at maximum capacity (a figure hard to obtain even with the best of settings) it would be 50 per cent of the total air supply at 10 per cent of maximum capacity (again assuming constant low excess air in the furnace). So, therefore, with 50 per cent primary air and 50 per cent furnace leakage there would be no secondary air, and, in order to obtain satisfactory combustion by the use of a small amount of secondary, the excess air would have to be increased.

Fig. 3 gives a chart by means of which the required quantity of tempering air may be readily determined. For example, suppose the primary air requirements of a pulverizer are as shown by Fig. 2, and that, owing to low velocity in the coal pipe to the burner, at 20 per cent pulverizer capacity, the tempered primary air cannot exceed 250 F without fire hazard. If the air from the preheater is 525 F then the required tempering air will be $0.60 \times 3.5 = 2.1$ lb per pound of coal or more than 15 per cent of the total combustion air required for an average coal when burned with about 30 per cent excess air.

Magnitude of Air Leakage as Determined by Test

Few tests on modern steam-generating units have been conducted to determine the magnitude of the air leakage or infiltration from all sources and over a wide range of evaporation rates. Consequently, most of the data available were obtained incidental to the determination of efficiency or capacity or in connection with special tests on particular parts of the unit such as, for example, the boiler or air heater only. Therefore, the inconsistency of the results of infiltration calculations made from such tests is not to be wondered at, in view of the high probability of errors in the measured data.[4]

It is also difficult to devise a scheme for correlating the air infiltration data obtained from different units. Probably no two installations will give exactly identical results, owing to the many factors affecting leakage which were discussed in Part I of this series. For this reason no fixed rule can be given and the allowance to be made in the design of any unit is necessarily a result of experience and good judgment. The *Cochrane Handbook*, however, makes the general statement that even in well-cared-for boiler settings, the in-leakage amounts to 25 per cent of the gases passing through the uptake. And the Bureau of Mines in an earlier publication[5] sets the figure at 30 per cent for the average plant.

Some very consistent test data were recently obtained from a modern steel-encased unit having a water-cooled furnace of 9000 cu ft, set under a straight-tube boiler of about 150,000 lb per hour maximum evaporative capacity. This unit was pulverized-fuel-fired by the direct method but no tempering air was used in connection with the pulverizers. Flue gas in the furnace was not analyzed, so the calculated total combustion air supply includes both furnace and boiler-setting infiltration. Neglecting the outward leakage from the hot air ducts, the difference between the weight of air passed through the air heater and the total weight calculated from boiler outlet CO_2 must be infiltration. It is plotted as per cent of the total combustion air in Fig. 4. Sufficient data to segregate infiltration from the three known sources were not taken. However, the pulverizer infiltration was appreciable at low rates of operation due to the fact that primary air flow was regulated by a damper at the pulverizer inlet. It is thought, therefore, that the dash lines of Fig. 4 indicate approximately the magnitude of leakage from the various sources.

Fig. 4 also shows the reduction in CO_2 in the flue gas at the boiler outlet that occurred at the low combustion rates. The reason for this was as explained in connection with the discussion of Fig. 2. During the low rating tests on this unit trouble was encountered due to fires in the fuel pipes and spilling of coal from the burners. To maintain a high velocity in the pipes it was necessary to use more primary air through the pulverizer than was necessary merely to sweep it free of powdered coal. The furnace infiltration also is partly to blame for the reduced

[4] See "Air Leakage in Steam Generating Units," Part II, April 1934.
[5] "Sampling and Analyzing Flue Gases," Bureau of Mines, Bulletin 97.

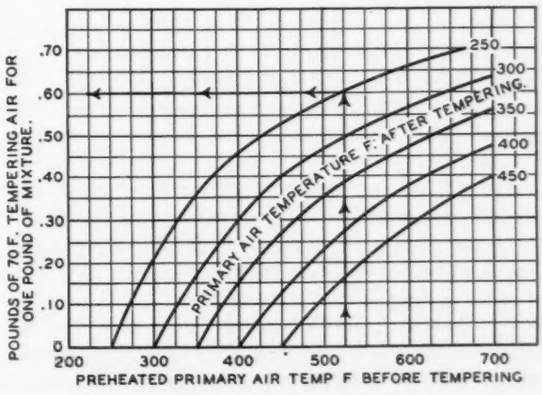


Fig. 3—Tempering air required for one pound of tempered mixture and for different preheated primary air temperatures

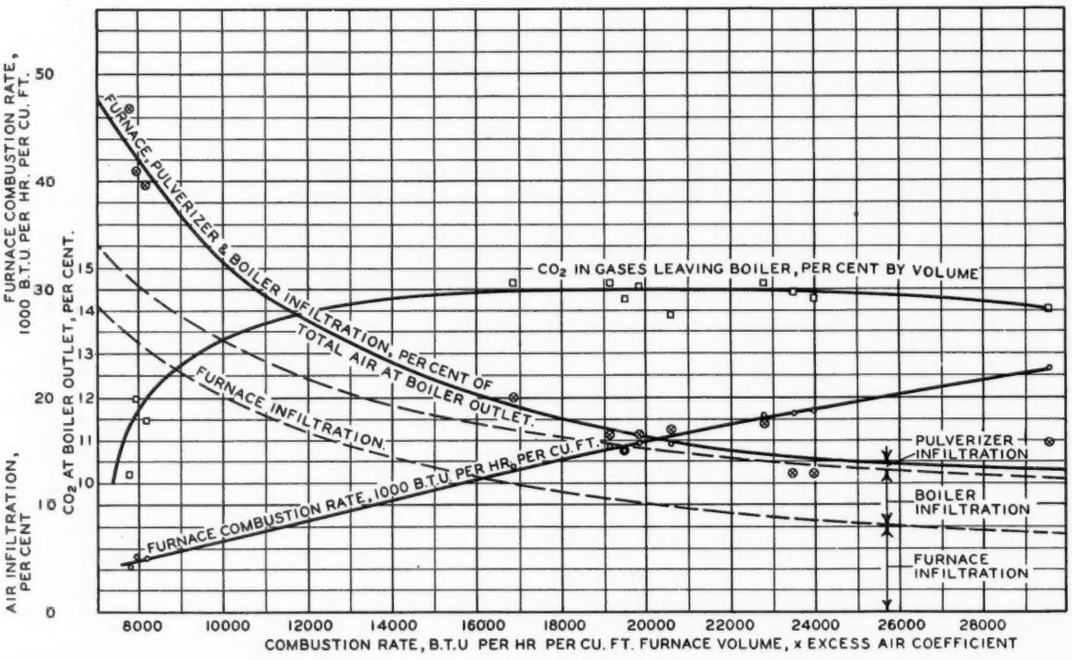

FURNACE COMBUSTION RATE, 1000 B.T.U PER HR. PER CU. FT.
AIR INFILTRATION, PER CENT
CO2 AT BOILER OUTLET, PER CENT.
CO2 IN GASES LEAVING BOILER, PER CENT BY VOLUME
FURNACE, PULVERIZER & BOILER INFILTRATION, PERCENT OF TOTAL AIR AT BOILER OUTLET.
FURNACE INFILTRATION.
FURNACE COMBUSTION RATE, 1000 B.T.U. PER HR. PER CU.FT.
PULVERIZER INFILTRATION
BOILER INFILTRATION
FURNACE INFILTRATION
COMBUSTION RATE, B.T.U PER HR PER CU. FT. FURNACE VOLUME, x EXCESS AIR COEFFICIENT

Fig. 4—Relation between combustion rate, infiltration and CO_2 as determined from tests of a typical direct-fired pulverized fuel steam generating unit. (Boiler and pulverizer infiltration segregated by assumption)

CO_2 because, had it been less, more secondary air could have been supplied without increasing the total air supply.

From numerous tests on oil, gas and pulverized-fuel-fired units having air heaters, sufficient judgment can be developed so that a manufacturer should be able to predict the total air supplied under control within plus or minus 10 per cent at low rates of evaporation and plus or minus 5 per cent at maximum rate. But unless the outward leakage of hot air from ducts and windbox can be eliminated, the air-heater heat balance will be of little value in estimating setting infiltration on stoker-fired units. Therefore, experience gained with other methods of firing must often be applied, due allowance being made for the higher combustion rates, per cubic foot of furnace volume, generally encountered in stoker firing.

The traveling-grate and chain-grate stoker furnace presents a special problem because of the various arch arrangements. When two arches are used the gas velocity through the throat may be high enough to cause considerable flow resistance. This will be overcome by the static draft created by the hot gases (stack effect). The result is that high negative pressures are not encountered in a properly operated furnace of this type and consequently if the arches are tight the furnace infiltration will be less than for the same size underfeed-stoker installation.

To make a general statement of the magnitude of leakage into boilers, economizers, air heaters and breeching would be impossible. Records of tests can be produced to show extremely large drops in CO_2 due to infiltration. The *Cochrane Handbook* cites two instances where by sealing brick boiler settings the CO_2 was increased 4.5 points and 3.1 points, respectively. The Bureau of Mines reports tests showing an increase in gas weight of 22.4 per cent between the end of the combustion chamber and the stack. The Bureau of Mines also reports tests[6] showing about 3 points CO_2 drop between boiler outlet and economizer outlet, which represents an increase in gas weight of about 25 per cent.

The only other item of importance is the outward leakage of air which may occur between the forced-draft fan and the furnace. Its magnitude will be negligible for steel ducts if the solid and flexible expansion joints between sections are properly constructed and sealed. Leakage from burner windboxes for oil, gas and pulverized fuel can readily be located and eliminated, but stoker windboxes present a different problem because of their size and the number of control rods for dampers, dump plates, distributing rams, etc., depending on the type of stoker. Concrete and masonry are used extensively for ducts and windboxes of stokers and these may cause considerable leakage due to invisible porosity. In connection with efficiency tests on a 2500 sq ft stoker-fired marine boiler[7] the Bureau of Mines measured the air handled by the forced-draft fan by means of an inlet orifice. It was found to be 12 to 20 per cent higher than the total air supply calculated from the analysis of gases leaving the boiler. In other words, the outward leakage was probably 22 to 30 per cent of the total air supply calculated at boiler outlet, assuming that furnace and boiler setting infiltration combined was 10 per cent. *Cochrane Handbook* gives a table of draft requirements for underfeed stokers in which 35 per cent leakage between fan outlet and fuel bed has been allowed for.

Summarizing on the subject of magnitude of leakage, it may be said that purchasers of new steam-generating units may specify and expect to get installations much tighter, all the way from ash-pit to stack, than indicated by some of the tests mentioned above. They can

[6] "Tests of a Large Boiler Fired With Powdered Coal," Bureau of Mines, Bulletin 237.
[7] "Tests of Marine Boilers," Bureau of Mines, Bulletin 214.

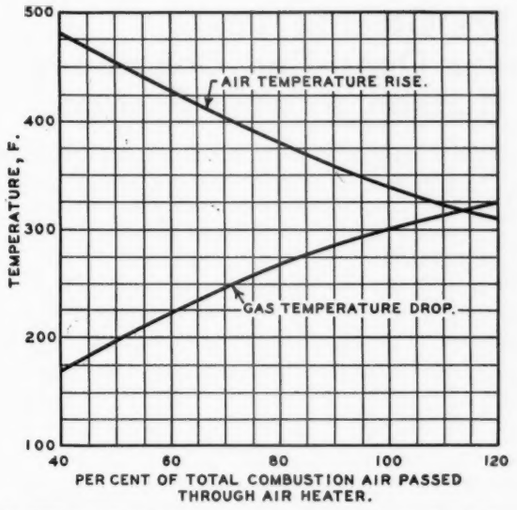


Fig. 5—Effect of quantity of air passed through air heater, on gas temperature drop and air temperature rise, based on constant entering temperature of both gas and air and constant gas quantity

reasonably expect not more than one point drop in CO_2 (about 7 per cent leakage) across a new brick boiler setting and less across a steel-encased boiler. They can purchase economizers and air heaters across each of which the CO_2 drop should not exceed one-half point. And they may specify multiple-retort stokers having not more than 5 per cent outward leakage from the windbox.

Economic Importance of Air Leakage

Air infiltration into a furnace operating at a high combustion rate is of little importance except when an air preheater is used in conjunction with the boiler. Its effect on the total air supply is negligible at high furnace combustion rates because most of the leakage air will combine with the fuel as effectively as if it were supplied under forced-draft control. For the same reason excessive air infiltration into a pulverizer is unimportant. And further, excessive outward leakage from the air duct or windbox is important only due to its effect on size and power consumption of the forced-draft fan. But if an air heater is used the conditions are different. The air recovers heat from the flue gas by means of heat transfer by conduction and convection, and because of the temperature difference or heat head between the two fluids. In-leakage of cold air to the furnace and pulverized-fuel system decreases the quantity of air passed through the air heater and, therefore, decreases both the heat transfer rate and the mean temperature difference between the two fluids. Conversely, outward leakage from duct or windbox improves the air heater performance and, superficially at least, counteracts the loss of economy due to the in-leakage.

Fig. 5 shows graphically the effect of varying only the quantity of air passed through an air heater receiving a constant quantity of gas at constant temperature. Gas temperature drop and air temperature rise are plotted against per cent of total combustion air. The heat absorbed by the air heater is proportional, almost directly, to the gas temperature drop. Therefore, for the average conditions for which Fig. 5 was plotted, 20 per cent reduction in the air quantity passed through the heater amounts to the equivalent of about 1 point loss in overall efficiency. And conversely, 20 per cent outward leakage of preheated air from a duct or windbox, if unaccounted for in a heat-balance test, would make the overall efficiency superficially high by about 1 point.

Air infiltration into a furnace operating at low combustion rates affects not only the performance of the air heater but also that of all the other equipment beyond the furnace such as boiler, economizer and induced-draft fan. Various reasons why it is generally necessary to increase the total air supply per pound of fuel burned were mentioned in connection with discussion of Figs. 2 and 4. It is a well-known fact that although increasing the excess air in the furnace has the effect of lowering the gas temperature entering the boiler, it nevertheless has the opposite effect on the gas temperature leaving the boiler. From tests conducted by the Detroit Edison Company[8] on a large stoker-fired boiler unit, operated at constant steam output with variable excess air, it was found that as the CO_2 in the gases was varied from 15 to 9 per cent the increase in flue gas temperature per point drop in CO_2 was 27 F between 15 and 13 per cent CO_2, and about 15 F between 11 and 10 per cent CO_2. The stack gas losses increased two points (2 per cent of efficiency) for each point decrease in CO_2 below 14 per cent.

Fig. 6 shows the calculated variation of boiler exit gas temperature due to change in excess air supply figured for an arbitrarily chosen boiler operating at high rating with entering gas temperature assumed constant and no CO. The calculations are based on heat transfer rates obtained by test and the variation in temperature shown by the curve is thought to be representative of average conditions, although different boilers will give different results depending on their size, efficiency and rate of evaporation.

If the gases leaving the boiler, for which Fig. 6 was plotted, are then passed through an economizer, without further increase in their weight due to infiltration at the boiler damper frame or into the connection to the economizer, it will be found that the variation in temperature due to excess air will be less as the gases leave the economizer. The greater the total heat absorption of the economizer the greater will be this reduction in gas temperature variation.

It can be shown, however, that the infiltration of air into the gas duct to an economizer or into the economizer

[8] "Effect of Excess Air on Flue Temperature and Efficiency," A. Kristian Bak, *Power*, Vol. 59, No. 17, 1924.

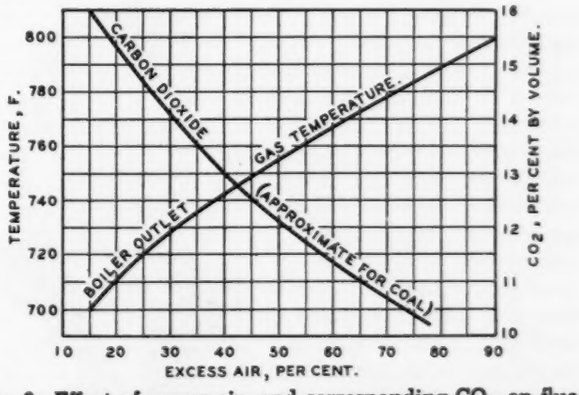


Fig. 6—Effect of excess air, and corresponding CO_2, on flue gas temperature leaving boiler; calculated for boiler of assumed size with constant entering gas temperature and constant fuel weight burned

itself will decrease its efficiency, and therefore the overall efficiency of the steam-generating unit, even though the temperature of the gases leaving the economizer will be less than if the leakage did not take place. The Bureau of Mines gives the heat balance for the economizer previously mentioned[9] and reports that the loss due to air leakage into the economizer was as high as 13 per cent of the heat supplied to the economizer and as high as 1.8 per cent of the heat in the coal fired. The gain due to stopping the leaks would have been even greater than the reported loss due to leakage, because the heat absorption by the water would have been increased.

The induced-draft fan size and operating cost also are greatly affected by infiltration. For example, 25 per cent increase in gas weight between furnace and fan may increase the required fan pressure over 50 per cent and the power consumption as much as 100 per cent.

In conclusion it is interesting to calculate what annual expenditure could be justified to keep the equipment tight assuming for example that a 10,000 sq ft boiler designed for a maximum capacity corresponding to 400 per cent of nominal rating is operated at a monthly load factor of 75 per cent and an annual use factor of 80 per cent. The loss in economy due to air leaks into a furnace and boiler setting is given in the *Cochrane Handbook* as 10 per cent of the heat generated in the furnace, as a conservative estimate for the average power plant. On the basis of 72 per cent average operating efficiency with 14,000 Btu coal costing $5. per ton delivered to the furnace, the expenditure of about $17,500 per year could be justified if the entire 10 per cent loss due to leakage could be eliminated; or conservatively assuming that only 5 per cent of the total heat generated could readily be recovered by eliminating part of the leakage, the expenditure of anything less than about $8,750 per year for this item would effect a saving in operating costs.

[9] "Tests of a Large Boiler Fired with Powdered Coal," Bureau of Mines, Bulletin 237.

H. S. Colby Made Sales Manager of Combustion Engineering Company, Inc.

The appointment of H. S. Colby as General Sales Manager of Combustion Engineering Company has just been announced by R. M. Gates, Vice President.

Mr. Colby, until recently, was President of the Air Preheater Corporation. His technical education was acquired at Cooper Union and Columbia University and for the past twenty years he has been engaged in the design, production and sales of steam generating equipment involving boilers, stokers, pulverizers, economizers and air preheaters. This background of experience and an extensive acquaintance in the field of steam generation, eminently fit him for his new duties, which include direct charge of district office activities.

Valve Closing Tests Under Open-End Conditions at Hudson Avenue Station

A VERY unique valve test was run at the Hudson Avenue Station of the Brooklyn Edison Company to determine the ability of a 600-lb, 750-deg, 12-in. motor-operated gate valve to close in event of a break in a steam line.

The sketch, Fig. 1, shows the schematic layout of the steam line in which the valve is located. The valve follows a 12-in. pressure-reducing valve in a tie line which connects the high- and the low-pressure sections of the station. Eight safety valves are on the low-pressure side of the tie line to protect the low-pressure system from over pressure. During the test the low-pressure section of the station was not operating and open end line conditions were obtained by blocking open the safety valves. The safety valve discharge is conducted to the boiler plant roof through a relief line.

Nine complete runs were made with steam flows ranging from 27,000 to 705,000 lb per hour, as determined by a Bailey flowmeter fitted with a fast clock. Motor current and voltage and valve stem travel were recorded during each run by synchronized recording meters. Pressure ahead of the valve versus valve travel was also taken by means of a steam-engine indicator. Pressure in the safety-valve relief line just above the hood and at the track floor was taken by recording gages.

The curves in Fig. 2 show the valve data obtained during the 705,000-lb per hour flow and Fig. 3 the safety-valve relief line pressure data.

The valve motor is rated at 104 amp, for 5 minutes at

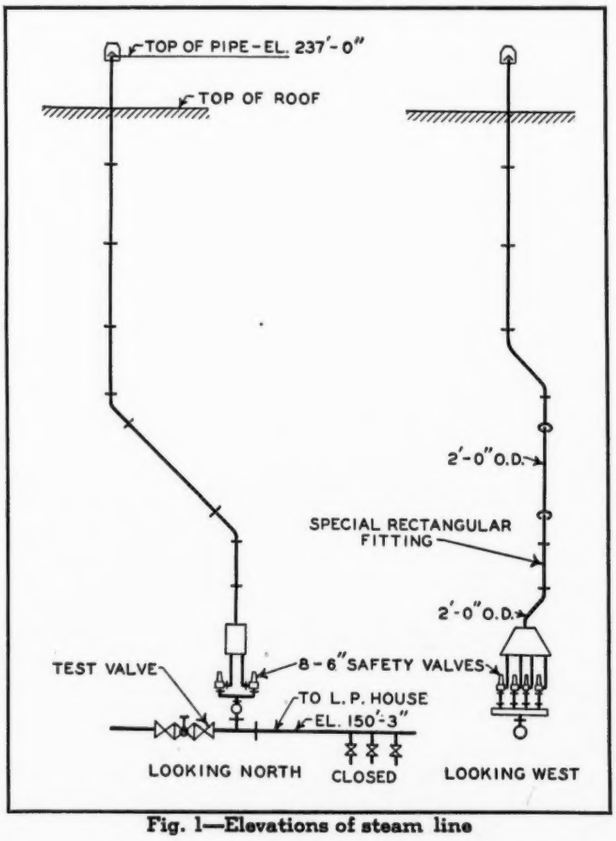


Fig. 1—Elevations of steam line

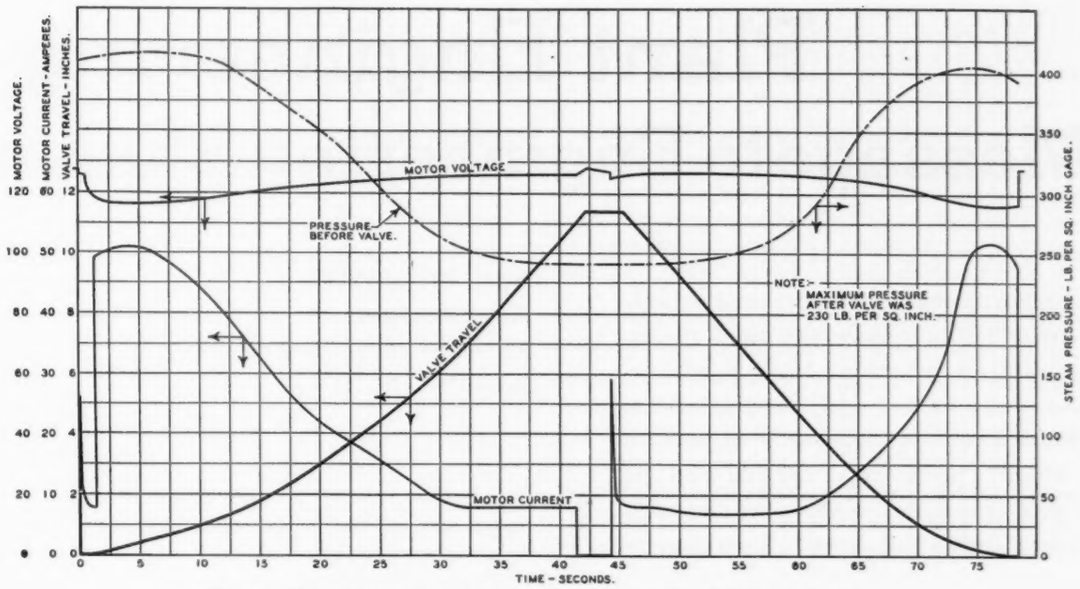

MOTOR VOLTAGE.
MOTOR CURRENT - AMPERES.
VALVE TRAVEL - INCHES.
STEAM PRESSURE - LB. PER SQ. INCH GAGE.
TIME - SECONDS.
MOTOR VOLTAGE
PRESSURE BEFORE VALVE.
VALVE TRAVEL
MOTOR CURRENT
NOTE:- MAXIMUM PRESSURE AFTER VALVE WAS 230 LB. PER SQ. INCH.

Fig. 2—Valve data obtained during test with 705,000 lb per hr flow

125 volts. It is evident from the motor-current curve that the motor capacity is ample to take care of the severe conditions of opening and closing on an open end line. The following tabulation of steam flow and motor current shows that under open-end line conditions the motor current is for practical purposes, independent of steam flow, and that maximum closing and opening currents are about the same.

Steam Flow Lb per Hour	Max. Opening Motor Current Amperes	Max. Closing Motor Current Amperes
27,000	45	33
140,000	46	47
160,000	43	35
200,000	48	41
255,000	40	43
460,000	48	50
705,000	51	52

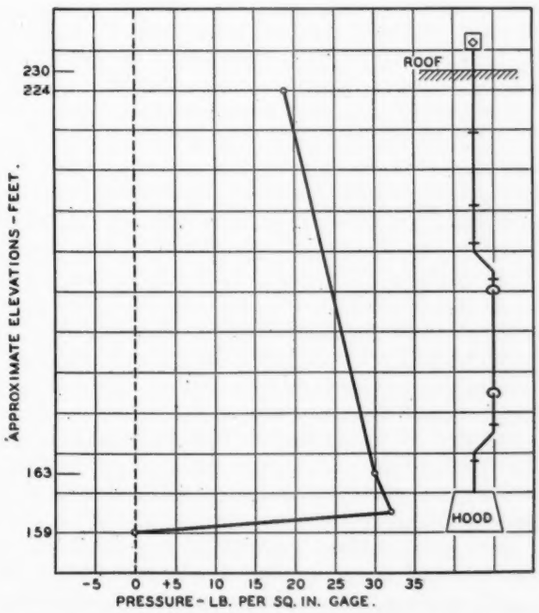


Fig. 3—Pressure distribution curve with 705,000 lb per hr flow

The time required to close the valve under the various steam flows was as follows:

Steam Flow Lb per Hour	Time Required to Close—Seconds
27,000	22
140,000	29
160,000	25
200,000	27
255,000	30
460,000	31
705,000	34

The pressures obtained in the safety-valve relief pipe as shown on the curve, Fig. 2, were higher than anticipated and were above the pressures selected for the line design so that after the tests some sections of the line which had flat surfaces had to be reinforced. During the 460,000 lb per hr run a pressure record was obtained at a point in the hood of the safety-valve relief line. This record showing a pressure of 16 lb was reached after the valve had been operating toward an open position for 25 sec, and remained approximately at this value until 60 sec had elapsed. The valve was fully closed after an elapsed time of 66 sec.

After the test the valve was dismantled for inspection and it was found that the Monel metal body seat ring and Monel metal wedge ring on the down stream side of the valve were badly galled. The valve was not opened for inspection prior to the test so that it is not definite that the galling was due to the severe test conditions but it is believed that this was the case.

An interesting note in connection with the test is that the noise of the steam escaping at the roof carried more than a mile to the company's main office building and could be heard in the engineering offices above the level of street noises which are high in the Metropolitan area.

Philip Sporn has recently been appointed vice president in charge of all engineering activities of the American Gas and Electric Co. A graduate of Columbia University, he joined this company in 1920 and held successively the positions of assistant to the electrical engineer, electrical engineer and chief engineer of the company.

Plant Engineering—
A Message to Engineering Students

The author, speaking from long experience in this field, outlines the steps necessary to gaining practical knowledge, reviews the many duties and responsibilities of the plant engineer, especially as pertain to the steam, power and service departments, and concludes with some observations as regards his status with the management.

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THE SELECTION of a branch of engineering as a vocation would be a difficult problem for an experienced engineer yet the student is compelled to make this decision at an age of inexperience. Because the opportunity for a wrong decision is so great, modern educators are recommending a curriculum of general engineering with the development of a specialty deferred until after graduation. That is good logic based on experience but it concerns only the acquisition of theories and sound engineering depends upon both theory and practice. Consequently practice or application must be considered.

The first decision to be made by the student is whether he will enter the branch of pure science or that of management. If the decision is for management, a broad knowledge of application is a necessary adjunct to a broad knowledge of theory and it is safe to say that plant engineering affords the greatest opportunity for wide experience in the practical. In that vocation, it will not take long to decide whether one is fitted for the pursuit of pure science or for the life of an executive. If it is the latter, he will be getting much first-hand information and training that will be most valuable in any specialty that may later develop.

Steps in Gaining Experience

In order to get a picture of the scope of experience let us follow the probable advancement of a student entering the employ of a large process plant. He may be started at drafting and later be placed in the plant particularly if he is not a very good draftsman. "Out in the plant" he will probably be started by following up operation and maintenance in the boiler and power house which means that at first he will be a high-class errand boy between the foreman and the plant engineer. At first, he seeks information and every one is most respectful but later when he has opinions of his own he clashes with the foreman and discovers that he has to learn how to disagree with the foreman and the men under him. Then he becomes more experienced, dares to disagree with the plant engineer and finds that the method of disagreeing with the foreman is not the one to use with a superior. Later he finds that more consultation and less dictation is better in both cases. Then he learns something about the men under the foreman, their various types, capacities and limitations. Then he begins to realize that human engineering enters very largely into the success of any organization.

When he has become proficient in the boiler and power plant he is moved from one department to another until he is capable of operating and maintaining the equipment and buildings of the entire plant. In material handling he has received and shipped materials in trucks, cars, barges or steamers. Material has been moved by coal towers, cranes, gas and electric trucks, belt conveyors, bucket conveyors and elevators, platform elevators, drag conveyors, scrolls, air conveyors and a great variety of pumps. Metallurgy has been met in all of its phases—tensile strength in heavy and light equipment, in various materials, at high temperatures and constant or variable loads—erosion with coal, cinders, metal on metal and flow of liquids and process materials —corrosion in boilers, economizers and flues, process acids, brine and moisture—machining and welding—protective coatings such as paint, galvanizing, sheridizing, electro-plating and metal spraying. Chemistry has been used in manufacturing processes, corrosion, sanitation, boiler feedwater and process water. Experience has been had in electric, mechanical, hydraulic and pneumatic regulating devices. Steam engineering consisted in its generation, transmission and use for power and heat. Electrical engineering was included in the generation, transmission and application of power, illumination and communication. Mechanical movements were involved in innumerable types of equipment from heavy to light automatic. Many types of lubrication have been encountered.

In addition to power he has had experience in such service commodities as compressed air, hot and cold water supplies, refrigeration, air conditioning and elevators. Then there were such activities as fire, explosion, theft and accident prevention and hygiene. He has encountered many municipal and state regulations in regard to labor, air and river pollution and licenses. In process work were mining, pulverizing, screening, drying, evaporation, both single and multiple effects

under vacuum with condensers, pumps and steam jets, conveying, sorting, cutting and many more too numerous to mention. Packaging, warehousing and shipping were also included.

It is true that he may not encounter all of the above mentioned items but he will experience most of them.

Construction and Alterations

After he has mastered operation and maintenance he will be put on major alterations, construction and erection of equipment and buildings. It may involve anything from an attachment on an automatic machine to a heavy bridge crane. Heavy machinery may have to be installed by taking it through an opening made in the side walls of a building or opening a hole in the roof and walls, girders and columns must not be overloaded. The sea wall may be bulging and must be tied back. Foundations may be sinking and have to be replaced or under pinned. A rearrangement of electric power cables or piping system may have to be installed. Concrete, tile, wooden and mastic floors must be laid. A new boiler, turbine-generator or switchboard must be installed. An old process must be removed and a new process with its particular machinery, conveyors, elevators, electrical work and transmission gear must be installed. A steel stack has corroded to the danger point or a brick chimney has been struck by lightning and must be made safe. Tests must be run to determine the effect of temperature, pressure or velocity in some one process. A sudden demand for one article of production necessitates a quick increase in production capacity. A major accident means quick resumption of production by temporary expedients.

A complete list of his activities would include many more items but the above is sufficient to indicate the variety of his work. During this period he has become experienced in laying out work for and supervising such artisans as mechanics, machinists, carpenters, pattern makers, masons, plasterers, pipe fitters, tinsmiths, copper smiths, blacksmiths, ironworkers, belt men, pipe coverers and riggers.

The next episode in the life of our young engineer is his promotion to the position of plant engineer. For the first time he sees his department as a whole and a salvo of demands for greater production, more efficiency and better continuity of service at less cost make him realize that at last he is a real executive. Heretofore he has had to think on his feet and make quick decisions. Now, although he must still mix occasionally in the excitement and make quick decisions, he finds it necessary to draw apart and spend long periods in deep meditation.

Organization work must now be given more intense study. There must be sufficient talent to cover every demand for research, testing, design, construction, operation and maintenance. The organization must be flexible in order to handle an unusual amount of work in any one classification. There must be the proper amount of supervision. All men must be properly trained and adapted to their particular jobs. Firehands, loafers, idlers, clowns and cantankerous minds must be reduced to a minimum. The wage scale should be comparable with that paid in the neighborhood for similar work and good workers attracted by humane treatment, aroused pride in one's work and advancement given voluntarily to the deserving. A runner-up must be trained for every key man. A full day's work but no more must be exacted from every employee.

Records Necessary

A system of records, that is as simple as possible and yet give a true and complete story, must be maintained. Total and unit figures of performance, efficiencies and costs must be quickly available, as well as a comprehensive log. A good storeroom system is also essential. It is important that any additions or reductions in physical value of the plant be accurately recorded for inventory accounts. The accounting system is usually determined by the accounting division but is usually well classified by production steps and not by equipment. The plant engineer usually finds it necessary to make his own classification of accounts and generally keeps a few unofficial records of his own. It is natural for the accounting department to have misconceptions of the work in the engineering department and the plant engineer can be of considerable assistance in making the records portray an accurate story. He is called upon to give an allocation to various departments of the cost of heat, light, power, compressed air, refrigeration and other services. The division of costs of any new construction and installations to various processes is the duty of the plant engineer. In addition to records of the past he must prepare budgets for the future.

Knowledge of Laws, Regulations and Safety Measures

One of his interesting obligations is the study of laws, rules and regulations of state, municipal and insurance company origin. He must have a thorough knowledge of the engineering involved in the prevention of accidents to employees, fire and explosions as well as to sanitary conditions in order that danger shall be minimized, regardless of laws. Many hazards exist that are not covered by rules or regulations. A thorough knowledge of building laws and licenses is necessary. A plant engineer can save considerable money for his company by consulting and cooperating with governing boards instead of opposing them and treating inspectors as a group of parasites. Many laws if carried out to the letter would cause undue hardship upon the owner and many exceptions will be made if the boards are convinced that the plant engineer is sincere in his effort to remove hazards.

Contact with Management

To the new plant engineer, organization, records, accounting and outside contacts seem to be a continuation of his previous experience as an assistant but now he has his first direct contact with the manager and it is an awesome sort of thing. He has heard and read much about handling men under him but no one ever coached him in handling men over him. His first experience is an increased regard for the management and the second is the need of a new vocabulary. Gradually he begins to study the personnel of the management to find out what they are trying to accomplish and in what manner they want it done. When he has accomplished that, he is in a position to help them obtain their objective by engineering methods unknown to them. Next the plant engineer finds that he must study the management from another angle. There are times when he sees the advantages of new processes or new equipment and he

must get the idea across to the management. Thus he becomes a salesman and he must be a good one.

The above-mentioned activities of the plant engineer must be well handled but the real success comes from ability to solve a wide variety of engineering problems. There will always be sufficient operating problems and enough problems brought up by the management to keep the plant engineer busy but there should be many problems developed by him. Conditions within a plant are always changing and the plant engineer is the one most capable of identifying the time when new processes or new equipment will increase production, increase quality of product or reduce costs. Furthermore, it is the duty of the plant engineer to have such records of past performances as to make possible the intelligent selection of equipment to meet the needs.

Handling the Power Problems

Probably the greatest loss that an engineer can cause his employer is in the handling of the generation of power, particularly in an industrial plant where a large quantity of steam is used in process. A boiler house is expected to last about 25 years. It represents a large investment and the fuel consumed during its life amounts to a much larger sum. The most common error made by a plant engineer is the lack of records showing the size and character of steam and electric loads. Whether it is lack of foresight or poor salesmanship in getting the management to realize the importance of meters, the blame is the engineer's. When the time arrives to rebuild the old boiler house or put in new generators or purchase power there should be complete records showing the relative power and heat demands during any one hour, during seasonal loads and during shut down periods. Without these records it is very easy to saddle a company not only with a high operating cost for a long period but actually to curtail production in the busy season. It will probably result in an unbalancing of the steam systems, overloading the exhaust steam system and blowing thousands of dollars worth of steam into the atmosphere. Under these conditions it is impossible to show a profit on the installation of equipment for increasing the thermal efficiency of the plant because the more heat that is saved, the more steam is wasted into the atmosphere. This is not only possible, it has happened more than once.

Change in raw materials or in manufactured articles, more efficient use of heat and constantly increasing electrical load are continually altering the heat balance but there are many ways of restoring the balance by new equipment such as heat exchangers or changing motor-driven units for turbine units, exhausting into the process system, instead of into the exhaust system. The plant engineer is solely responsible for controlling the heat balance.

Naturally, a plant engineer cannot be a highly developed expert in all of the lines that he has to handle. Neither can he engage consulting engineers on every problem that arises. Consequently he will try to excel in two or three specialties and have sufficient working knowledge of the other lines to solve most of their problems. In judging his own ability to solve a problem without outside help, he cannot afford to be either optimistic or pessimistic.

The possibilities for advancement are quite broad. If he is a good executive, he may be made manager of the plant or he may be made the executive engineer of a corporation with several plants. He may enter the employ of a consulting engineer or establish himself as a consulting engineer. The research engineer of one of our greatest public utility corporations secured his first experience as a plant engineer.

In his own company he is held in high regard, usually ranking next to the manager, and in addition to being called upon for engineering advice is sometimes called into executive conferences. By outside engineers, who come in contact with his work, he is respected, but to others in the engineering profession he is too often considered a high grade master mechanic. However, the employment of technical plant engineers by industrial concerns is increasing and plant engineers are making themselves more prominent in writing for technical magazines so that their standing will rise.

The requirements of a successful plant engineer are a wide experience in both theory and practice, executive ability and a disposition and physique for hard work. One thing that should be borne in mind is that, during the first years of experience, the graduate must not neglect the technical side even though little of it will be required of him at that time. The cadet engineer should look at the plant as a whole and study it from the technical standpoint. He should join engineering societies, subscribe to technical magazines and attend technical meetings. The most helpful agency is that of contact with engineers of one's own age.

Activities of A.S.T.M. on Refractories

As a result of recent work of A.S.T.M. Committee C-8 on Refractories, new proposed specifications will be recommended for approval as A.S.T.M. tentative standards covering widely used refractory materials.

The tentative method of control test for resistance of fire-clay brick to thermal spalling action, often termed the "water dip spalling test," is to be withdrawn. The committee is considering other methods to determine resistance of refractory materials to thermal and structural spalling. The withdrawal of the Definitions of Terms Relating to Heat Transmission of Refractories will also be recommended. In place of these, the committee proposes the approval as tentative of the "Symbols for Heat Transmission" as approved by the American Standards Association.

A number of changes are also to be proposed as tentative for inclusion in the Methods of Testing High-Temperature Heat Insulation.

Several studies and research projects are being carried on by the committee. The investigation on methods for determining thermal conductivity of refractory materials is to be continued; a panel spalling test is being studied; and work is contemplated on load tests of heavy duty fire brick at temperatures higher than those now specified in the standard test.

The report of Committee C-8, to be presented at the A.S.T.M. annual meeting in Atlantic City, June 25–29, will review the year's work and give in detail all of the recommendations, proposed standards, etc., involving the refractories field.

Anthracite as a Filter Medium

Solution of Silica from Sand and from Anthracite in Hot, Alkaline Waters Compared

By

H. G. TURNER* and G. S. SCOTT**

Parallel studies of the solubility of silica from sand and from anthracite in 1.0 per cent and 0.1 per cent sodium-hydroxide solutions at 2, 6 and 10 atmospheres were made. Alkaline waters at high temperatures contain free alkali, which dissolves silica from a siliceous filter medium when filtered through a pressure filter containing such a medium, and deposit this silica as a hard scale in a boiler using the filtered water. The comparative advantages of the two filter media are brought out by this study, which shows that solution of silica from sand takes place indefinitely and at a fairly rapid rate, while a very small quantity of silica is dissolved from anthracite at a comparatively slow rate.

CONSIDERABLE interest has been shown of late in the mechanical properties of anthracite as a filter material. Anthracite is essentially non-siliceous, and it was therefore felt that a study of the relative amounts of silica dissolved from sand and from anthracite should prove of decided value at this time.

It has been shown that sodium carbonate under boiler conditions decomposes into sodium hydroxide and carbon dioxide.[1, 2] Joos states[3] that sodium carbonate, or soda ash, present in boiler waters (either naturally or as the result of chemical treatment by soda ash, boiler compound or a lime soda softener) undergoes a transition under steam conditions. He found a 72-per cent conversion of sodium carbonate to caustic soda at 145 lb per sq in. pressure, and 82-per cent conversion at 280 lb. The conversion at 145 lb was complete in about two hours, and was but little affected by rating.

Straub and Larsen's results are in good agreement with those of Joos. Their work covered the range from 147 C to 243 C. They found, in addition, that the dynamic equilibrium set up in the boiler practically coincided with the true equilibrium.

It thus becomes apparent that solution of silica by carbonate waters at elevated temperatures is due principally to the free alkali content of those waters. It was therefore decided to restrict the tests to the ability of sodium-hydroxide solutions to dissolve silica from sand and from anthracite instead of going through the additional lengthy series with alkali-carbonate solutions.

Experimental Work

Two series of experiments were run. In the first, 100 cc of 20 × 30-mesh (U. S. screen) sand was placed in an autoclave and one liter of sodium-hydroxide solution added. The autoclave was then closed and the burner lit. It was maintained at the desired pressure for a definite period of time, then cooled and the silica content of the solution determined by the standard analytical procedure. A parallel series using 20 × 30-mesh anthracite instead of sand was carried on. Fresh samples of sand and of anthracite and new solutions of sodium hydroxide were used for each test.

In the second series, 100 cc of sand was placed in the autoclave and one liter of one-per cent sodium-hydroxide solution added. The charge was maintained at 10 atmospheres for $3^1/_2$ hr. After cooling, the silica content of the resulting solution was determined. This same sample of sand was then weighed and placed in the autoclave with a new solution of sodium hydroxide and another test made. This procedure was repeated a number of times. A parallel series using anthracite instead of sand was also carried on.

A blank run was made at 10 atmospheres, by going through the full procedure, but without addition of either sand or anthracite. The result obtained (22 ppm of silica) was subtracted from all determinations.

Discussion of Results—Sand

Reaction Velocity.—The data for sand obtained from the first series of experiments are shown in Table 1, and plotted as Figs. 1 and 2. The last two columns of Table 1 show the rate constants, calculated according to the equation[4]

$$K = \frac{2.303}{O.t} \log \frac{s}{s - c} \qquad (1)$$

where K is the rate constant,

O is the sand surface in square centimeters,

t is the contact time in hours,

s is the theoretical weight of silica per liter of NaOH which would be dissolved in infinite time, based on the assumption that the compound formed is $Na_2O-1.42SiO_2$, and is expressed in grams per liter.

c is the actual weight of silica per liter of NaOH which is dissolved in time t, in grams per liter.

Several ratios of silica to soda are given for sodium silicate.[5] For sodium silicate formed under the condi-

* Director of Research for the Anthracite Institute.
** Assistant Director.
[1] Paul: Boiler Chemistry, page 38 (Longmans, 1919).
[2] Straub and Larsen, *Ind and Eng. Chem.*, 24/1416 (1932).
[3] *Power*, Nov. 12, 1929.
[4] Getman, "Outlines of Theoretical Chemistry," page 377, (Wiley, 1918), Noyes and Sherrill, "Chemical Principles," page 139 (Macmillan, 1929); Taylor, "A Treatise on Physical Chemistry," Vol. 2, page 1030 (Van Nostrand, 1931).
[5] Vail, "Soluble Silicates in Industry" (Chem. Cat. Co., 1928).

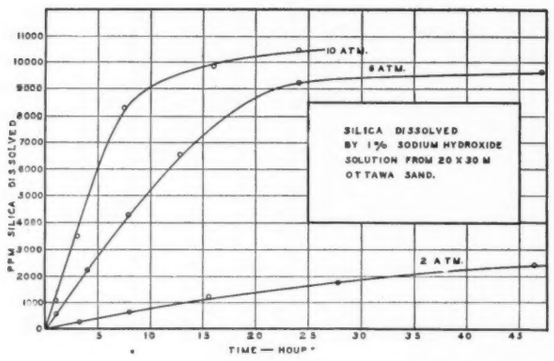

10 ATM.
6 ATM.
2 ATM.
SILICA DISSOLVED BY 1% SODIUM HYDROXIDE SOLUTION FROM 20 X 30 M OTTAWA SAND.
PPM SILICA DISSOLVED
TIME — HOURS

Figure 1.

tions of the first series of tests, the formula is approximately

$$Na_2O - 1.42SiO_2$$

This statement is based on the fact that this ratio appears to fit the results best. Based on the above formula, the numerical values of s are 10.65 × per cent NaOH (using initial value).

Rough measurements of the sodium silicate adsorbed by the sand indicate an end-product corresponding to

TABLE NO. 1

SILICA DISSOLVED BY SODIUM HYDROXIDE SOLUTIONS FROM 20 x 30 MESH OTTAWA SAND.

TEMP. °F.	TIME HOURS	INITIAL CONCENTRATION % NaOH	FINAL CONCENTRATION p.p.m. Silica	K x 10^5	K x 10^5 (Ave.)
365	1.00	1.01	1080	2.28	
	3.00	0.98	3383	2.17	
	7.53	1.03	8518	2.47	
	15.95	1.07	10528	1.91	
	24.00	1.01	10568	2.86	2.34
324	1.01	1.04	566	1.17	
	4.11	1.00	2002	0.95	
	7.73	1.03	4368	1.02	
	12.82	1.01	6588	1.00	
	24.00	1.01	9303	1.00	
	47.07	1.04	10018	0.55*	1.03
270	3.35	1.00	263	0.181	
	8.10	0.98	578	0.158	
	15.63	1.02	1273	0.164	
	27.87	0.98	1698	0.125	
	46.67	0.99	2428	0.101	0.146
358	1.00	0.099	158	3.355	
	3.55	0.098	306	2.41	
	6.73	0.102	398	1.59	
	16.65	0.100	623	1.20	
	23.25	0.097	767	1.30	2.09
324	1.00	0.098	92	2.29	
	3.83	0.100	165	1.07	
	7.65	0.101	246	0.82	
	15.17	0.102	299	0.67	
	24.03	0.099	361	0.54	1.08
270	1.00	0.100	36	0.86*	
	6.00	0.100	78	0.31	
	6.43	0.099	78	0.30	
	16.32	0.100	140	0.21	
	24.67	0.100	158	0.16	0.28

* Omitted from average.

$2N_2O - 3SiO_2$. The data presented, however, refer to the silica content of the solution.

When first calculated, the values of K, when plotted against time, showed a maximum for each series of 1-per cent solutions. The reason for this appeared when the results for the second series of tests were plotted (see Fig. 6). This chart indicates an increase in surface with decreasing weight up to a maximum, followed by a gradual decrease to zero. The original figures were based on a constant surface area of 5100 sq cm for the 100-cc quantities of sand, on the assumption made by Hixson and Crowell[6] that the surface area decreases linearly with the weight loss, and for weight losses of the order 0–3 per cent could be neglected.

Fig. 6 indicates that the assumption made by Hixson and Crowell is not applicable to this case, and data taken by Fieldner, Hall and Galloway[7] indicate that the present results do not constitute an isolated case of this type, since those authors show a similar relationship between weight loss of carbon during activation and increase of surface area, the surface area being measured relatively by iodine adsorption.

Values for the sand surface area, O, used in the above formula for reaction velocity constants were as follows:

From the second series of experiments, tabulated in Table 3, values for s were calculated from the analyses of the NaOH solutions, values for c were determined analytically, and $\log \frac{s}{s-c}$ computed. These computed figures were then multiplied by $\frac{2.303}{3.5\,K}$ to obtain the corresponding surface areas. The value of K to be used as a base was calculated from the one-hour test at 10 atmospheres and 1-per cent NaOH from the first series, using the area of the new sand as calculated geometrically on the basis of a spherical shape. Inspection of Fig. 8 will show that this assumption of a spherical shape was by no means a wild one at zero time. There was a mean weight loss of 0.34 per cent in this one-hour test, and the surface areas computed by this procedure tend, therefore, to be slightly low due to the assumption of 4000 sq cm as the surface area at zero time, obtained by extrapolation, and to be slightly in error due to the slight departure from a spherical shape. Inasmuch as we are more interested in the relative surface areas, these small errors due to basic assumptions decrease to within the experimental error and may be neglected.

In Fig. 7 the surface areas are plotted against the mean weight loss of the sand. The smooth curve, drawn through the points, follows the equation

$$O^2 = 16 + 19.75\,w - 0.3975\,w^2 + 0.001985\,w^3. \qquad (2)$$

where O is the surface area in square centimeters, and w is mean weight loss in per cent of original weight.

Equation (2) was used to compute the values of O

[6] *Industrial and Engineering Chemistry*, 23/923 (1931).

[7] U. S. Bureau of Mines, Tech. Paper 479, page 18 (1930).

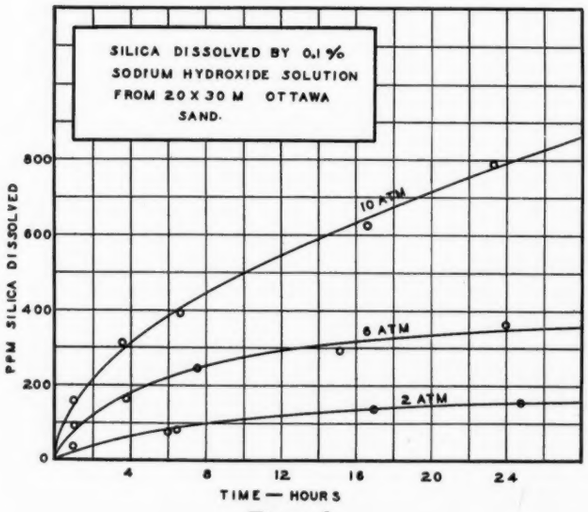

SILICA DISSOLVED BY 0.1% SODIUM HYDROXIDE SOLUTION FROM 20 X 30 M OTTAWA SAND.
10 ATM
6 ATM
2 ATM
PPM SILICA DISSOLVED
TIME — HOURS

Figure 2.

TABLE NO. 2.

SILICA DISSOLVED BY SODIUM HYDROXIDE SOLUTIONS FROM 20 x 30 MESH ANTHRACITE.

TEMP. °F.	TOME HOURS	INITIAL CONCENTRATION % NaOH	FINAL CONCENTRATION P.P.M. Silica
365	1.00	1.00	342
	3.01	1.05	555
	7.55	1.06	688
	23.83	1.02	688
324	1.00	1.02	194
	5.51	1.01	434
	14.92	1.03	558
	24.10	1.04	583
270	4.37	1.00	222
	15.05	1.00	402
	24.07	1.00	440
365	1.00	0.100	61
	3.07	0.100	110
	7.53	0.102	135
	15.95	0.100	174
	24.33	0.099	176
324	1.00	0.101	32
	8.87	0.101	81
	16.82	0.099	125
	24.15	0.100	151
270	3.03	0.099	33
	7.30	0.100	55
	17.73	0.101	69
255	24.00	0.100	69

used in Equation (1) for the calculation of velocity constants. These constants are shown in the last two columns of Table 1.

It will be noted from examination of Table 1 that the velocity constants decrease with time where the quantities of silica dissolved are low, and that the relative decrease of velocity constants is greater as the dissolved silica is less. This is taken to mean that there exists a small quantity of fine silica dust in the sample. (0.03 to 0.05 per cent would be sufficient to account for the observed variation.) It has been suggested by Baer[8] that a very thin layer of hydrated silica may exist on the surface of the sand grains, and be sufficient to account for the observed

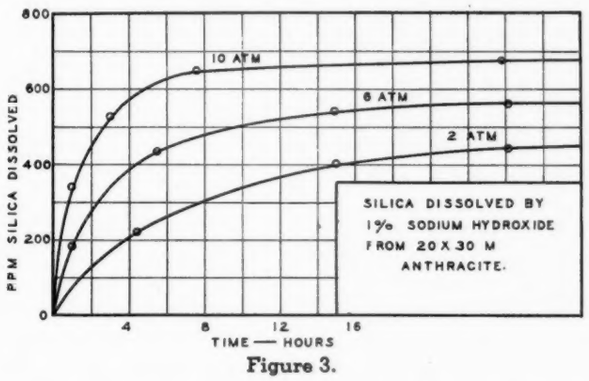


Figure 3.

variation. The writers incline toward the dust idea because a similar phenomenon apparently takes place with the anthracite also and will be mentioned later.

The last column of Table 1, and Fig. 5 shows the variation of the reaction velocity constant for sand with temperature. Since we have data for three points from each concentration of sodium hydroxide, we are in position to use a three-constant equation of the form[9]

$$\ln K = -\frac{A}{T} + BT + C \qquad (3)$$

From the data for 1-per cent solutions, the constants become, from simultaneous equations, $A = 73{,}600$; $B = -0.0933$; $C = 155.47$.

From the data for 0.1 per cent solutions, the constants become $A = 22{,}600$; $B = 0.01496$; $C = 29.09$.

If we average the two sets of constants, we obtain the equation

$$2.303 \log K = -\frac{48{,}100}{T} - 0.541\,T + 92.28 \qquad (4)$$

[8] Orally, George Baer, Asst. Prof. Ceramics, Pa. State Coll.
[9] Taylor, "Treatise on Physical Chemistry," Vol. 2, page 969 (Van Nostrand, 1931).

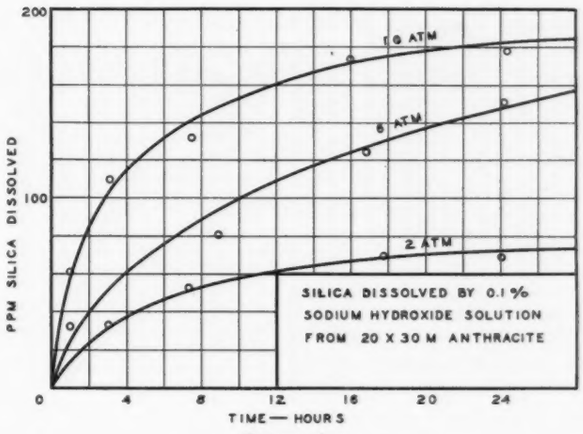


Figure 4.

which describes the variation of reaction velocity with temperature. The smooth curve in Fig. 5 is a plot of this equation, and the circled points represent the experimental data from which the equation was calculated. The apparently high result for the 0.1-per cent solution is doubtless due to ability of pure water to dissolve silica.

From inspection of this curve, it is apparent that diffusion is the controlling factor, at least over most of the range covered. We must therefore expect that the velocity of the liquid relative to the sand surface will be of great importance in determining the rate at which sodium-hydroxide solutions will dissolve silica sand.

Effect of Alkali Treatment on Sand

During the second series of experiments, the total loss of weight of the sand was 64.7 per cent. Fig. 8 illustrates the appearance of the sand grains at the end of this series of experiments. The sand was not only differentially attacked as shown in the photograph, but also reduced in size. A sizing test of this sample showed

Screen Opening, mm	Per Cent Passing
0.074	0.10
0.149	0.47
0.250	1.77
0.42	9.27
0.50	41.35
0.59	82.88

which gives an effective size of 0.42 mm and a uniformity coefficient of 1.31 as against an original effective size of 0.61 mm and uniformity coefficient of 1.21.

The total loss of sand by direct weighing, shown in Table 3 (107.6 grams) is larger than the total silica taken into solution (102.26 grams), as shown by the analytical data in Table 3, because some of the finest

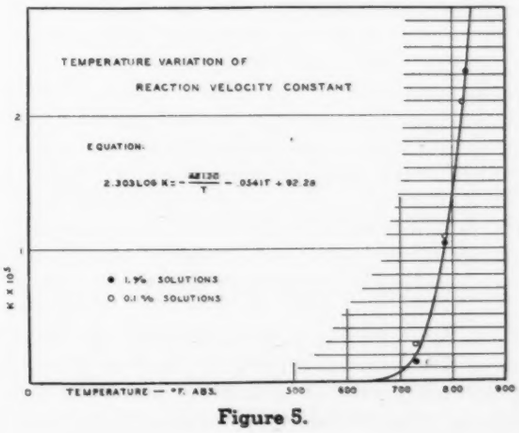


Figure 5.

particles were lost during the washing of the sand by decantation and in the several handlings of the sample, as well as by loss of some of the adsorbed sodium silicate during the several washings.

TABLE NO. 3.

TEST NO.	OTTAWA SAND WT.AT START	% NaOH	P.P.M. Silica	ANTHRACITE WT.AT START	% NaOH	P.P.M. Silica
0	162.1	1.00	5288	70.3	1.01	547
1	156.2	1.01	5688	68.6	0.99	358
2	149.7	0.97	6513	69.2	0.99	238
3	142.1	0.98	6913	67.4	0.97	176
4	134.9	0.98	7288	66.7	0.97	123
5	127.2	1.00	7418	66.9	0.98	101
6	119.9	1.04	8138	66.3	1.03	92
7	111.3	1.01	7763	66.6	1.03	78
8	103.5	1.03	3513*	66.5	1.01	68
9	99.7	1.02	8093	66.5	1.01	49
10	91.3	1.02	7568	66.2	1.00	50
11	83.5	1.00	7133	67.9	1.00	34
12	76.0	1.02	7248	67.3	1.02	28
13	68.4	1.01	6938	66.1	1.02	20
14	61.3	1.04	6758	65.7		
	54.5					
TOTAL SILICA			102260			1962
TOTAL LOSS IN WEIGHT,	107.6			4.6		

* After several attempts to repair leaky gaskets, the run was discontinued before the time was up.

Discussion of Results—Anthracite

Reaction Velocity.—Table 2 and Figs. 3 and 4 show the quantities of silica dissolved during varying periods of time from fresh, 20 × 30-mesh anthracite. During the first hour the rate at which silica is dissolved from sand is, in general, approximately three times greater than the rate at which it is dissolved from anthracite. Immediately after the reaction starts, the rate of solution from sand increases, due to increase of total surface as previously shown, while the rate of solution from anthracite decreases and ultimately reaches zero. Under these circumstances there appears to be no practical use for calculated velocity data. Any computations beyond the data shown in Table 2 and in Figs. 3 and 4 would simply be a volume of useless figures.

TABLE NO. 4.

TEST NO. (n)	CONC. OF SILICA P.P.M. (c)*	e^{kn}	k
0	542	----	----
1	362	1.496	0.402
2	240	2.258	0.407
3	181	2.993	0.365
4	127	4.267	0.362
5	103	5.260	0.332
6	89	6.085	0.299
7	76	7.13	0.281
8	67	8.09	0.261
9	49	11.05	0.267
10	50	10.73	0.237
11	34	15.94	0.251
12	27	20.07	0.250
13	19	28.51	0.258
8-13 incl.		(average)	0.254
0-7 incl.	1713**	(total)	

* The concentrations of silica have been corrected to 1% NaOH by direct proportion

** This total is the total of actual weights (from Table No.3) and not of the weights adjusted to 1% NaOH as above (Table No.4).

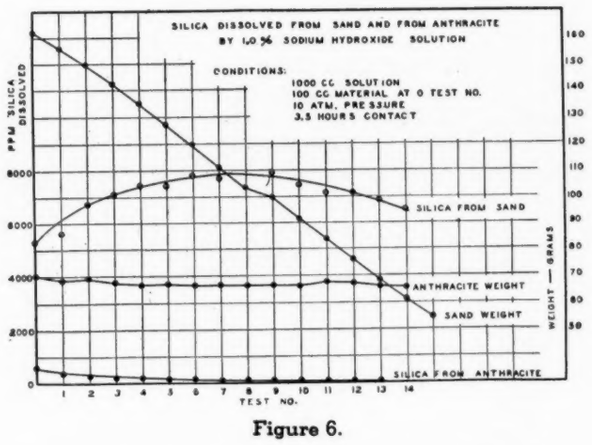


Figure 6.

On the other hand, the total quantity of silica that a hot, alkaline solution can possibly dissolve out of anthracite is of some significance. We note from Table 3 that the silica removed from anthracite becomes less with each successive treatment, or test. The underlying relation, from visual examination, is apparently exponential. Table 4 is set up on this assumption, the data used in the calculations being taken from Table 3. The exponential equation is of the form

$$C = C_o e^{-kn} \qquad (5)$$

In this equation, 542 is substituted for C_o and the various values of Column 2 are successively substituted for C. The ratio, C_o/C, is equal to e^{kn}, from which kn, then k, are readily obtainable from the "log-log" slide rule.

From this tabulation we may note fairly constant values of k from Test 8 to Test 13, inclusive. Using the average of Tests 8 to 13, we obtain the exponential equation

$$C = 542\, e^{-0.254\, n}, \qquad (6)$$

which describes the manner in which silica is dissolved from anthracite under the conditions of the second series of tests.

It will be seen, however, that Tests 0 to 7 show higher silica contents than called for by the exponential equation which fits the later results. The writers believe this to be due to the presence of fine dust in the original sample, and to correlate with the higher initial reaction rates for sand in those cases where the dissolved silica quantities are low.

We may calculate the total silica dissolvable, as the sum of the actual analytical weights of silica for Tests 1 to 7, inclusive, (which are higher than the exponential

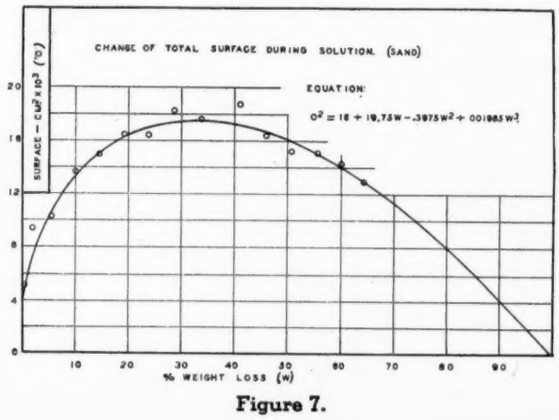


Figure 7.

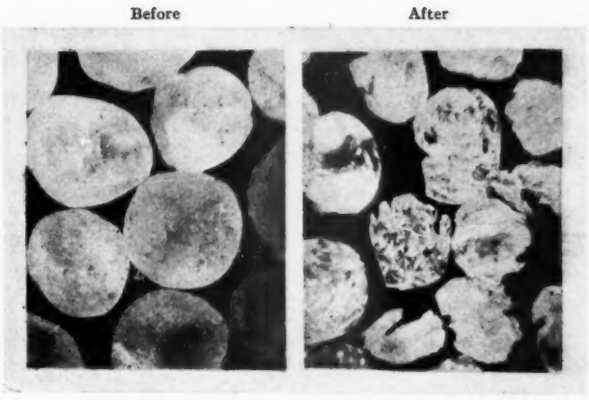


Fig. 8—Appearance of sand before and after sodium-hydroxide treatment

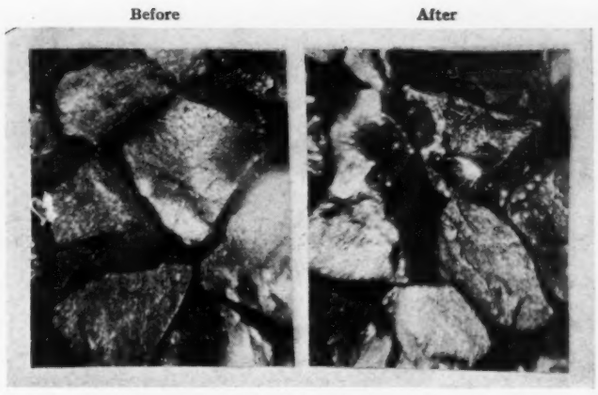


Fig. 9—Appearance of anthracite before and after sodium-hydroxide treatment

equation line), plus the sum of all values of C from $n = 8$ to $n =$ infinity (which corresponds to the exponential equation). The split is made between Test 7 and Test 8 because up to Test 7 the results must be added directly, while from Test 8 to infinity the sum of the various terms may be calculated from the exponential equation. Thus,

$$\text{Total Silica} = \sum_{n=0}^{n=7} C + \sum_{n=8}^{n=\infty} C$$

$$= 1713 + \sum_{n=8}^{n=\infty} C \qquad (7)$$

$$= 1713 + \sum_{n=0}^{n=\infty} C - \sum_{n=0}^{n=7} C$$

$$= 317 + \sum_{n=0}^{n=\infty} C \qquad (8)$$

$$= 317 + \sum_{n=0}^{n=\infty} \frac{542}{e^{.254n}} \qquad (9)$$

$$= 31\,7 + 542\left(1 + \frac{1}{e^{.254}} + \frac{1}{(e^{.254})^2} + \frac{1}{(e^{.254})^3} + \ldots\ldots\right) \qquad (10)$$

$$= 317 + 542\left(1 + \frac{1}{e^{.254} - 1}\right)^{*} = 2732 \text{ or } 2.732 \text{ grams} \qquad (11)$$

* Since

$$\frac{1}{x-1} = \frac{1}{x} + \frac{1}{x^2} + \frac{1}{x^3} + \ldots\ldots$$

This quantity, 2.732 grams, refers to 70.3 grams of coal. Since the original fresh coal contained 3.136 grams of silica by analysis, we may conclude that 0.404 grams, or about 13 per cent, is beyond the reach of the alkali, either completely protected by carbon, or in the form of minerals undecomposable by alkali at the temperatures used. Recent experiments in this laboratory with acid leaching removes about 87 per cent of the total ash, which, lacking further evidence, would seem to indicate that the carbon exerted a protective action.

The 65.7 grams of anthracite remaining after the second series of tests was burned to ash in a muffle and the ash analyzed. A 70.3-gram sample of the original fresh anthracite was treated in a similar manner. The results follow.

	Fresh Coal	Used Coal
Total weight of residue	5.8404 grams	2.2436 grams
Analysis of Residue, per cent		
Silica	53.70	51.20
Iron and Aluminium Oxides	40.30	41.28
Lime	1.20	0.44
Magnesia	0.44	0.16
Undetermined	4.36	6.92

From these data, we may calculate a weight balance.

	Fresh Coal	Used Coal	Material Dissolved
Silica	3.1363	1.1487	1.9876
Iron and Aluminium Oxides	2.3537	0.9261	1.4276
Lime	0.0701	0.0099	0.0602
Magnesia	0.0257	0.0036	0.0221
Undetermined	0.2546	0.1552	0.0994
TOTALS	5.8404	2.2436	3.5969

It is interesting to note:

(1) Silica from ash analyses..... 1.9876 grams
Silica from test analyses..... 1.962 grams
(Table 3)

(2) Loss of weight from ash analyses.. 3.6 grams
Loss of weight by direct weighing 4.6 grams
(Table 3)

The greater loss of weight by direct weighing is due to loss of anthracite from the many handling operations and the washing of the residue after each test. This is entirely analogous to the greater weight loss of sand obtained by direct weighing.

Application of Results

The results of these tests have a direct bearing on the filtration of hot, alkaline solutions or waters through pressure filters. They show that hot sodium-hydroxide solutions will dissolve sand completely, whereas these same solutions will dissolve only about 87 per cent of the mineral matter from anthracite and leave the carbon unaffected. After the initial loss, amounting to about 5 per cent of the total weight of anthracite, there is no further solvent action on the coal. This would indicate that anthracite could serve very effectively as a filter medium. Other properties of anthracite indicate that the use of anthracite for this purpose is not only feasible, but efficient and economical as well.

The writers wish to acknowledge helpful criticism from Mr. A. S. Behrman of the International Filter Co. and the cooperation of the School of Mineral Industries of The Pennsylvania State College.

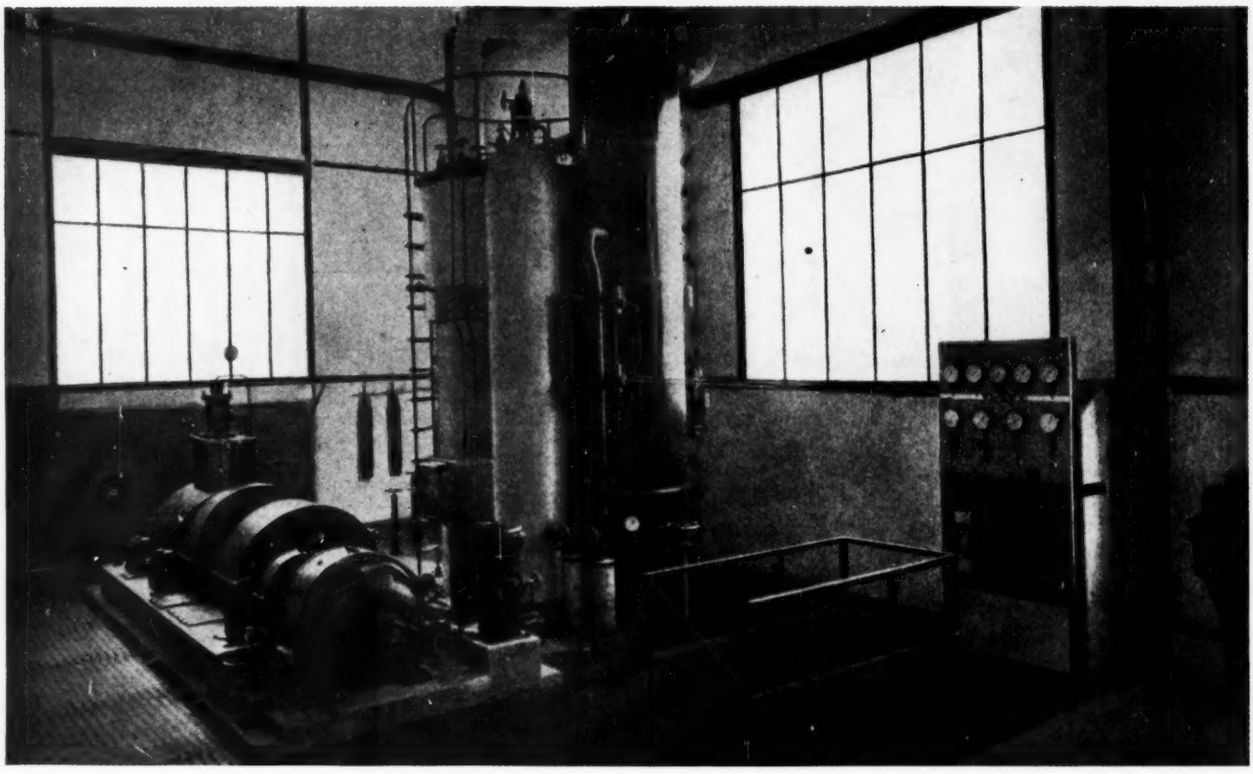


Fig. 1—View of 70,000 lb per hr Velox Steam Generator unit showing compact arrangement

The Velox Steam Generator*

By ADOLPH MEYER

Brown Boveri Steam Turbine Works, Baden, Switzerland

A description and technical discussion of the Velox steam generator in which the author deals with performance and includes test data. Heat release of nearly 900,000 Btu per sq ft per hr in the combustion chamber with a corresponding efficiency of 90 per cent and an evaporation of 101.5 lb per sq ft of heating surface per hour are noted. One feature of the boiler is that steam pressure and capacity can be attained in less than five minutes after starting up, and variations from half to full load can be met in about 20 sec.

THE Velox boiler differs essentially from most boilers which have been in use up to now. Its name, "Velox," refers to the high velocity of the flue gases, which are used to obtain exceptionally high heat transmission. The main features of this new boiler are as follows:

1. Combustion takes place under a pressure far in excess of that generally employed.

2. Partial transformation of the pressure into velocity, in order to obtain high flue gas speed. This speed may reach the velocity of sound.

3. The use of a turbo-blower, worked by an exhaust gas turbine, in order to produce the pressure in the combustion chamber referred to in (1) above.

4. Disposition of the gas turbine between two heat-absorbing parts of the boiler, in order to reduce the temperature of the flue gases so that cooling of the turbine is unnecessary, but also permitting of sufficient heat being left in the gas to provide the necessary energy for driving the turbo blower. The heat which remains is passed on through the feedwater heater section or sections of the boiler following the gas turbine.

5. A special device for separating steam from water which consists of a centrifugal separator without moving parts, in which the mixture of steam and water is injected tangentially on to the cylindrical wall of a container at such a velocity that it rotates with sufficient force to separate the steam and water by centrifugal action.

*From a paper presented before the Institute of Fuel, London, Eng., April 25, 1934.

6. The very small weght of steel and water taking part in the heat transmission which, together with the entire absence of fire bricks and the small storage of water which is necessary, allows for rapid starting of the boiler and gives very flexible operation under variations of load.

7. Forced circulation in the evaporating part of the boiler.

8. The control is entirely automatic.

9. Small weight.

10. Small space requirement.

11. High efficiencies over large variations in load.

These advantages have been obtained by the process of "supercharging," which is common practice in diesel work.

The advantages to be gained from supercharging are even greater when applied to a boiler, for, in addition to reduction in the size of the combustion chamber, the size of the heat transmitting system is reduced twofold; both by higher pressure as well as by higher speed of the flue gases.

In determining the practical range of velocities to be employed in the Velox boiler, it was soon found that to compensate for the higher expenditure on pressure-tight combustion chambers, and the pressure-producing machinery, a radical departure from accepted practice was necessary and velocities ranging from 900 ft per sec to 450 ft per sec have been adopted for the evaporator tubes: lower but still unusually high, velocities being used in feedheater and superheater. All these parts have therefore to be built "streamlined" so as to reduce friction losses as much as possible.

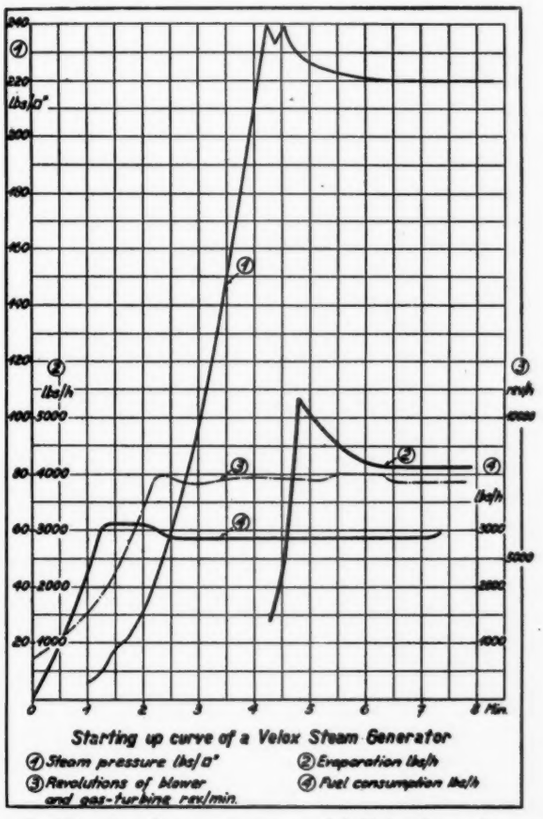

Starting up curve of a Velox Steam Generator
① Steam pressure lbs/□"
② Evaporation lbs/h
③ Revolutions of blower and gas-turbine rev/min
④ Fuel consumption lbs/h

Fig. 2—Showing how pressure and full load can be met in five minutes when starting up

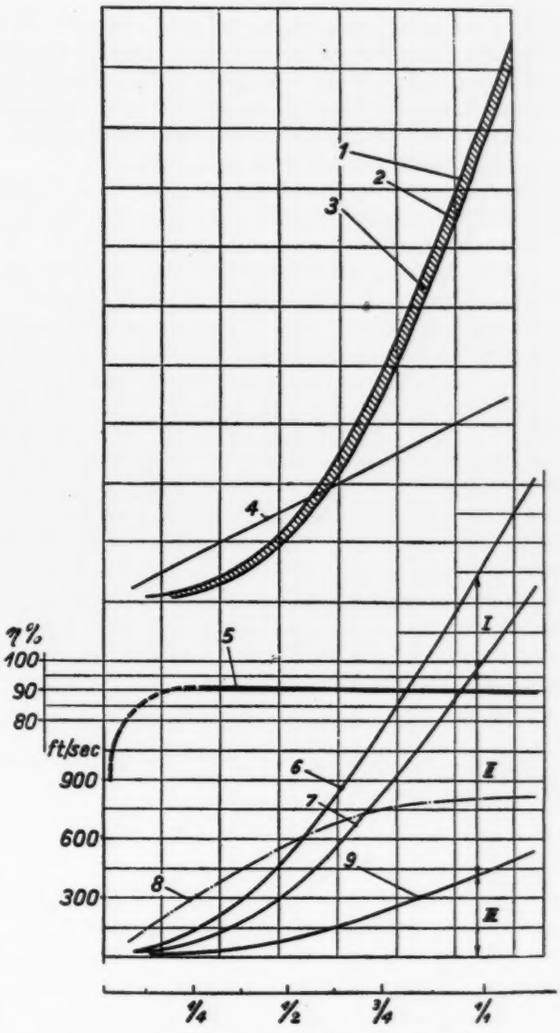


Fig. 3—Curves showing power requirements

Description of Velox Steam Generator

Fig. 4 represents a steam generator of the Velox type, and the diagram shows the temperatures and pressures prevailing in its different parts. The combustion of the fuel takes place in the combustion chamber *2* where air and fuel enter through the burner *1*, the air under a pressure of about 35 lb per sq in abs, the fuel under about 300 lb gage. The gases give up part of their heat content by radiation through the external walls of the evaparator tubes *3*, which line the wall of the combustion chamber. More heat is transmitted by convection, while the gases pass upward, through the internal tubes *3a* of the evaporators to the exhaust flue gas collecting chamber. Thus the initial temperature of combustion is reduced to about 1500 F, while the pressure drops to about 33 lb abs. With this temperature and under this pressure the gases enter the superheater *5*, to leave it cooled down to about 900 F (480 C) at a pressure of about 31 lb. The gas turbine *6*, which is then entered, causes the flue gas temperature to drop to about 700 F (370 C) while the pressure drops to about 16.5 lb abs. The corresponding heat drop is, apart from very small radiation and bearing losses, entirely converted into mechanical energy and

transmitted to the blower where it is reconverted into heat with a corresponding rise in the air temperature. Finally, the gases escape through the feedwater heater *7* which forms part of the chimney. From thence they continue, through the chimney *8*, to the atmosphere, where they leave cooled down to about 200 F (94 C).

The water and steam circuit is as follows:

The make-up water is fed by the feed pump *12* through the preheater *7* to the separator *4*, where it mixes with the evaporating water. This water is kept in continuous circulation by the circulating pump *11*, which pumps it through the combustion chamber 2 and evaporating tubes *3* back to the separator *4*, at the rate of about ten times the full load evaporation.

The water enters the separator through the nozzle *9* tangentially to its wall. This nozzle has the smallest sectional area of any tube in the circuit so that a sufficiently high pressure builds up in the tube before the water reaches the nozzle. This pressure is converted into velocity, and the resultant mixture of water and steam spins round on the inside of the cylinder. The water then falls through a small gap in the lower partition of the separator, while the steam is pressed out from the centre and enters the superheater *5*, where it is superheated up to the desired amount for use in a steam turbine.

Construction

Apart from the scheme itself, several details also show quite new features and therefore may be of some interest. Reverting once more to the combustion chamber, the evaporator tubes line the wall so as to protect it from radiant heat. They are bolted to the wall at the outlet end, and are guided so as to expand freely on the inlet end. Thus they can be removed from the chamber with ease. The inner tubes, which have nozzle-shaped inlets, show a remarkable diffusing action. They tend to re-transform kinetic energy into pressure, due to the rapid cooling of the passing gases with a corresponding reduction in volume. The flue gases pass the superheater elements in a longitudinal direction, the tube heads being streamlined so as to reduce pressure losses.

For the supercharging set the overall efficiency is of great importance. For the first boiler an impulse turbine-driven centrifugal blower was used, similar to those used for supercharging diesel engines. These boilers are now fitted with reaction turbines having three to four stages and axial blowers which have been specially developed for this purpose, the overall efficiency being of the order of 83 (turbine) × 73 (blower) = 60 per cent. With a view to reduction in weight, the gas turbine casing is entirely welded, while the blower casing is built up of a blade carrying a cast-iron body and welded air chambers. Both machines have heavy shafts so as to run below the critical speed, because their speed varies proportionately with the load, and they must be able to run continuously at any given speed between no load and full load.

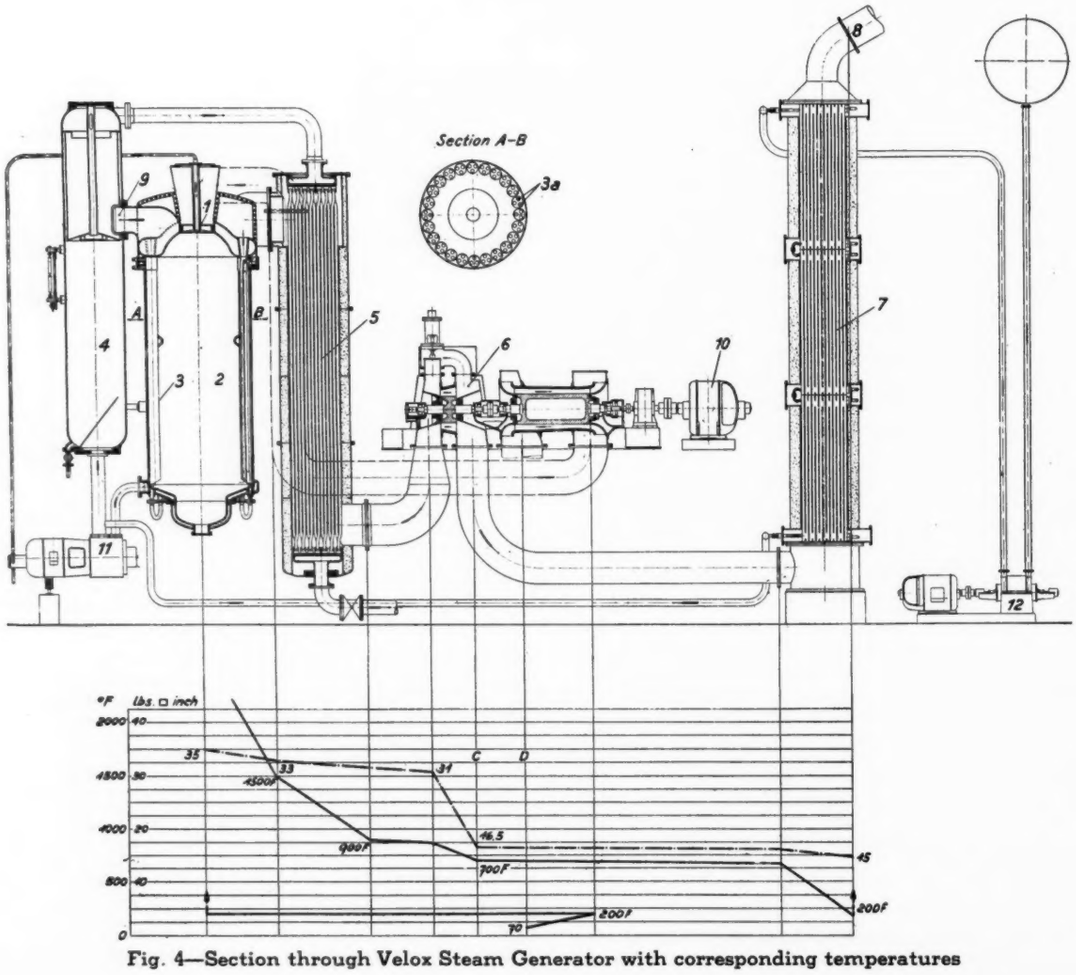


Fig. 4—Section through Velox Steam Generator with corresponding temperatures

The gases flow along the tubes of the preheater in the same manner as they do in the superheater. The elements of the preheater are therefore of the same design as those of the superheater.

The automatic governing system, in principle, is similar to that employed by Messrs. Brown Boveri on their turbines for the last 20 years.

Performance

Owing to the small amount of material taking part in heat transmission, and to the small water storage capacity which has been purposely kept down to a minimum, the boiler can be set to work in an unusually short space of time. Fig. 2 shows how steam pressure, temperature and quantity can be raised to full load conditions in less than five minutes. The same applies to load variations, which are so rapidly dealt with by the automatic governing system, that variations from half to full load and vice versa take only about 20 sec. Even when full load is cut out suddenly, no blowing off occurs.

As a Velox boiler is more a kind of steam-producing machine than a boiler, the whole unit can easily be erected and tested at the manufacturer's works. This has allowed tests to be made under normal working conditions, of which the following are available. Test No. 1 was made on an oil-fired works boiler built for 25,000 lb per hr capacity operating at 425 lb abs and a steam temperature of 750 F.

TABLE I—TEST RESULTS OF 10 TONS PER HOUR VELOX STEAM GENERATOR: BADEN WORKS

Combustion chamber volume	37 cu ft
Evaporator heating surface	244 sq ft
Superheater heating surface	285 sq ft
Preheater heating surface	710 sq ft
Total heating surface	1239 sq ft
Steam pressure	398 lb per sq in. abs
Steam temperature	716 F
Feed temperature	186 F
Steam generated	24,800 lb per hr
Heat content of steam less heat in feedwater	30,200,000 Btu per hr
Heat content of fuel	32,900,000 Btu per hr
Exhaust gas temperature	320 F
Efficiency, approx.	90 per cent
Heat liberated in comb. chamber	890,000 Btu per sq ft per hr
Steam generated—	
Evaporator H. surface	101.5 lb per sq ft per hr
Total heating surface	20.5 lb per sq ft per hr

The test results are given in Table I and are also represented by Fig. 3, where curve 1 shows the power required by the compressor, curve 2 power from the gas-turbine, while the difference (shaded) represents the additional power introduced by the auxiliary motor; curve 4 shows the speed variation of the turbo-blower set; curve 5 the efficiency; 6 the pressure at the discharge of the air compressor; 7 the pressure before the gas turbine; and 9 the pressure after the gas turbine, before preheater and chimney. Curve 5 shows the efficiency to be about constant over a very wide range of load.

Tests were also carried out at the works, on another boiler. The steam generator, with its governing system, is shown in Fig. 4 erected on its working site. This illustration gives a very good idea of the remarkably small space requirements of such a boiler. The boiler was constructed for a capacity of 70,000 lb per hr; a steam pressure of 300 lb per sq in. and a steam temperature of 700 F.

Particulars of this boiler are as follows:

Total evaporator heating surface	747 sq ft
Superheater heating surface	587 sq ft
Economizer heating surface	1722 sq ft
Total	3056 sq ft
Combustion chamber volume	173 cu ft
Steam production on test	69,700 lb per hr
Evaporation from evaporator heating surface	93.3 lb per sq ft
Evaporation from total heating surface	22.8 lb per sq ft

The boiler efficiency, including allowance for the power absorbed by all auxiliaries exclusive of the feed pump, was 90.85 per cent. The exhaust pipe, being relatively long and of insufficient diameter, caused the back pressure to be 22 in. water gage instead of 2 in. for which pressure it was originally designed and operated on site. Under normal working conditions the efficiency will approach 92.5 per cent in place of 90.85 per cent achieved.

In stationary plants its use is somewhat limited owing to the necessity for using liquid or gaseous fuel, but tests which have been made in the Baden works of Messrs. Brown Boveri show that the adaptation of the Velox steam generator to pulverized fuel may be possible in the near future.

[The paper concludes with a lengthy discussion and numerous diagrams showing the special adaptability of the Velox boiler to the merchant marine and naval work wherein large savings in both weight and space are possible.—EDITOR.]

Program of A.S.M.E. Meeting at Denver, June 25 to 28

The Semi-Annual Meeting of the Society will be held in Denver, Colo., June 25 to 28, with headquarters at the Cosmopolitan Hotel. The tentative program arranges for the meeting to open on Monday morning with two papers in the first session, one on passenger tramways and the other on smoke problems. The afternoon is given over to inspection trips. In the evening there will be a series of lectures with motion pictures giving a very complete story of the remarkable work of Boulder Dam.

Two simultaneous sessions will be held on the morning of the second day, one on heat engineering, covering air conditioning, experiences in the burning of pulverized lignite and petroleum coke. The other sessions will be an informal round-table discussion arranged by the Management Division on the subject of the decentralization of industry.

On Tuesday afternoon various plant inspection trips have been scheduled as well as another round-table discussion under the auspices of the Management Division, the topic of which will be the first year of industrial management under the NRA.

A semi-formal banquet with an address by the President of the Society, Col. Paul Doty, and a very unusual address by Capt. Pat O'Hay is to be held on Tuesday evening.

The papers to be delivered on Wednesday morning will include a discussion of beet-sugar manufacturing, mining in Colorado and a description of a unique example of cooperative manufacturing. The papers at another session, meeting simultaneously, will cover recent progress of high-speed light-weight railroad trains, with papers by representatives of the Burlington and the Union Pacific Lines.

Prof. Melvin L. Enger, a member of the University of Illinois faculty since 1907 and head of its department of Theoretical and Applied Mechanics since 1926, has been made Dean of the College of Engineering and Director of the Engineering Experiment Station of the University.

Developments in the Burning of Iowa Coal*

By JOHN M. DRABELLE
Mechanical and Electrical Engineer,
Iowa Electric Light & Power Co.

A study of draft losses with different arrangements of baffling in both old and new boiler installations at the Boone and the Perry Stations of the Iowa Electric Light & Power Company. Comparative test data are included.

IN 1930 we began to investigate draft losses through the various passes of boilers. Some things were encountered which were not explained on the basis of temperature drop in the gases as they pass over the heat absorbing surface of the boiler. Fig. 1 is a set of curves from some experimental research work of the General Electric Company, on a 600-hp B & W water-tube boiler with vertical baffling. This shows the range of temperatures of the gases entering the tube bank in relation to the tube number, numbering tubes from bottom to top. It will be noted from the top curve that in the first pass, gases enter the tube bank at a temperature of approximately 2180 F, and leave the top tube at approximately 1000 F. The gases then pass over and through the superheater, dropping to 940 F, and then begin their travel downward through the second pass (middle curve) of the boiler until they return to tube No. 1 at a temperature of approximately 640 F. The bottom curve shows the tube temperature variation in the last pass of the boiler, which shows that the last pass does but little work, the gases being reheated approximately from 570 to 590 F at the exit point.

TABLE I—TYPICAL ANALYSES OF IOWA COAL

Rated As Constituents	Good	Fair	Bad	Fair Perry Test
Moisture, %	7.88	12.36	14.42	14.81
Volatile, %	38.83	37.65	29.29	N. D.
Fixed Carbon, %	34.42	31.94	30.00	N. D.
Ash, %	16.23	18.06	26.29	18.2
Sulphur, %	4.75	5.75	7.00	4.85
Btu commercial	10,476	9712	7118	9219
Btu dry basis	11,774	11,081	8,316	10,932

N. D.—Not Determined.

TABLE II—AREAS OF PASSES IN BOILER AT BOONE

Arrangements	1st Pass—Area in Sq Ft	2nd Pass—Area in Sq Ft	3rd Pass Area in Sq Ft
Original vertical baffles (Design of 1912) (Fig. 2)	42	29	32
1st inclined baffling (Designs of 1924) (Fig. 3)	44	32.5	26
Modified inclined (Baffle design of 1930) (Fig. 5)	36	39	26
Perry (Design of 1933) (Fig. 6)	40.9	42.9	29.1

Table II gives the cross-sectional area of the first, second and third pass of an Edgemore boiler at our Boone, Iowa, station as shown in Figs. 2, 3, 4, 5, the last, Fig. 6, being the new installation of B & W boilers at Perry.

In pass No. 1, of the early design, we had a draft loss of approximately 0.01 in., in the second pass almost 0.2 in., or 20 times as much. The inclined baffle, as shown in Fig. 3, was designed on the basis of a temperature-volume reduction idea. These very low draft losses made for low velocities, and velocity means heat transfer. Consequently, as the work progressed at Boone we decided to move the first pass baffle forward, as shown in Fig. 5, and it will be noted from Table III, that we then increased this draft loss from 0.01 to 0.08 in., and in the Perry design (Fig. 6), by further decreasing the first pass area we increased this draft loss to 0.25 in., or 25 times as much as in the design of 1912. Also in Table III and Fig. 6 the draft losses through the boiler at Perry are more nearly uniform on a per pass basis than in the designs of 1930 or 1924, and the results obtained showed the soundness of this reasoning.

TABLE III—DRAFT LOSSES WITH THREE DESIGNS

	Design of 1924		Design of 1930		Design of 1933	
Pass No.	Area Sq Ft	Draft Loss In.	Area Sq Ft	Draft Loss In.	Area Sq Ft	Draft Loss In.
One	44	0.01	36	0.08	40.9	0.25
Two	32.5	0.19	39	0.165	42.9	0.24
Three	26	0.05	26	0.09	29.1	0.18
Boiler rating when readings were taken				190%	200%	210%
Hp of boiler used				410	410	500

Fig. 2 shows the early original design at Boone as of 1912, consisting of a small stoker, limited furnace volume, the arch down so low that it almost puts the fire out and straight vertical baffles with the old proportions which had been used for many years. This furnace, while it gave good, clean combustion and very little smoke, had the fundamental and very serious defect that very little coal per square foot per hour could be burned and the boiler developed only about 125 per cent of rating.

Fig. 3 was our first effort as of 1920 with the vertically inclined baffle. The furnace remained the same in volume and the arch and stoker were not changed. This gave us somewhat improved combustion conditions and a slight increase in rating.

Fig. 4 shows the change as of 1924; in this case the furnace volume was radically increased by almost 33 per cent, the arch was raised off of the fire and the stoker area was increased 15 sq ft. With this arrangement, considerably improved combustion conditions were obtained in respect to burning more coal per square foot per hour and developing higher ratings. We could depend upon 180 per cent, and at times get 200 per cent. This furnace, due to lack of turbulence and volume, was

* From a paper before the Iowa Engineering Society at Iowa State College, Ames, Ia., March 8, 1934.

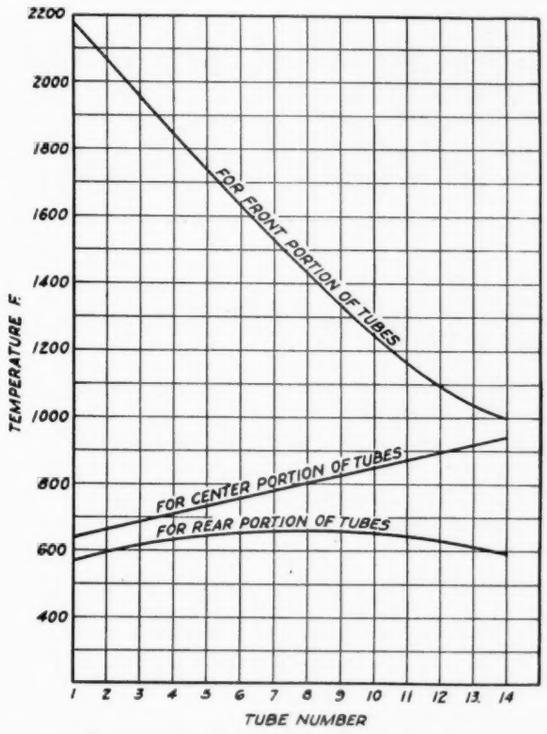


Fig. 1—Temperatures at different tube locations from tests at G. E. Co.

smokier than the designs of 1912 and 1920, and had the very serious objection that under certain conditions noisy combustion or drumming would take place.

This trouble led to a study of the first pass areas as shown in Tables II and III. We moved the inclined baffles forward, as shown in Fig. 5, so as to speed up the velocity of gas through the first pass, to increase the area of the second pass which, in the early designs, was suffering from rather high draft losses and attempted to get away from noise. This noise reduction was not fully realized until we opened the top of the boiler and put a flat construction across the drums so as to change the volume of the entire boiler setting, acting as a resonating chamber. The only explanation that we have for this drumming effect is that the high volatile gases burning in the furnace, have an action similar to the singing flame of hydrogen gas burning in a glass tube. This drumming effect at times was so violent that it could be heard several blocks from the power station and was very objectionable. The pitch of the sound was equivalent to the deep pedal note of a pipe organ. Mr. Wallace, our chief engineer at Boone, had noticed that if the fireman ran a slice bar into the furnace and disturbed the fuel bed, that this drumming would cease. He then began some experimental work, putting some plows or rakes on the stoker feed gate, which plowed grooves into the fuel bed and permitted more oxygen at the distillation zone. This, together with the changes in the resonating chamber of the boiler, stopped all drumming and noise.

TABLE IV—IMPROVEMENTS IN EVAPORATION WITH CHANGES

	Boiler Rating Hp	Grate Area	Furnace Value	Boiler Rating %	Av. Evap.
Design of 1912	410	81 Sq ft	246 Cu ft	125	5.5 lb
Design of 1920	410	81 Sq ft	259 Cu ft	150	6.0 lb
Design of 1924	410	96 Sq ft	409 Cu ft	180	6.35 lb
Design of 1931	410	96 Sq ft	386 Cu ft	200	6.50 lb
Design of 1933	500	126 Sq ft	1835 Cu ft	250	6.70 lb

Table IV, is a statistical history of the changes made and approximate improvement in evaporation obtained.

The above are average results taken from the power station records of the past.

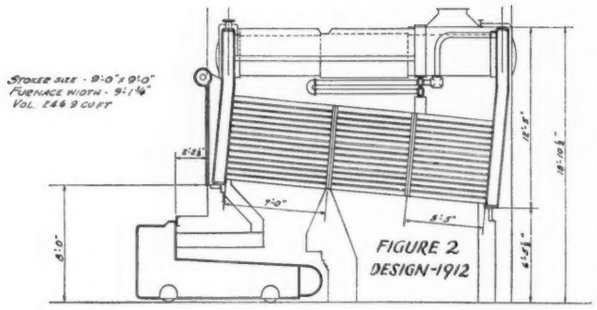
Fig. 6 is a cross-section of the new installation at Perry, Iowa, consisting of two B & W boilers fired by Green natural draft chain grate stokers having an active area of 126 sq ft. It will be noted that the boilers are set reversed, that there is a very large furnace volume, that there is plenty of reflecting refractory surface to maintain high furnace temperatures, and thereby secure more complete and cleaner combustion. There is a high angle arch, and an opening through the curved portion of the arch immediately in front of the stoker gate permits the injection of over-fire air.

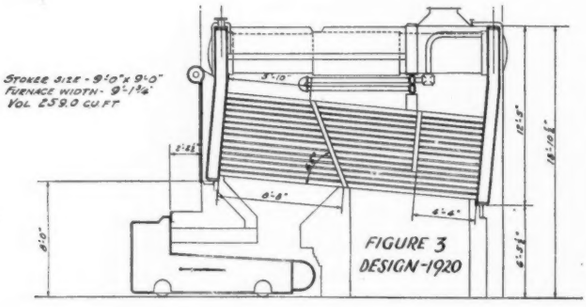
Referring now to Table III, it will be seen that the first-pass draft loss is 25 times that of the old design in Boone in 1924. The second-pass draft loss almost equals the first-pass draft loss, and is 0.05 in., or 25 per cent higher than Boone. Also the third-pass draft loss is almost four times as great as that at Boone in 1924 and is by 0.06 in. lower than the second-pass draft loss. The first-, second- and third-pass cross-sectional areas are greater. Theoretically, at least, the first pass should be the one of the greatest area. However, it is less than the second pass. The draft losses of passes one and two are approximately equal.

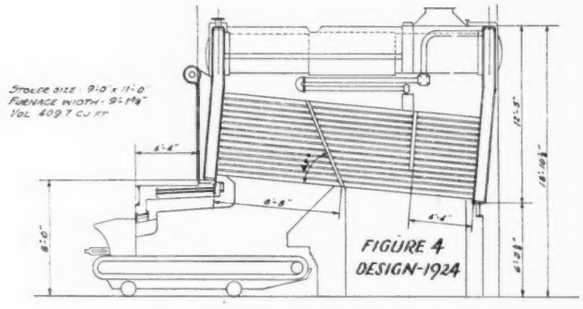
This installation has been in service but a short time. A series of tests, unfortunately of only three hours' duration each at various ratings, were run. The re-

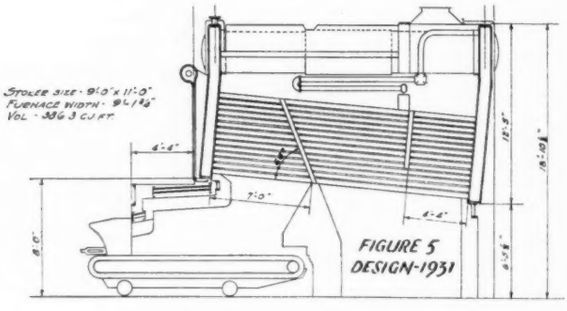
TABLE V—BOILER TEST DATA

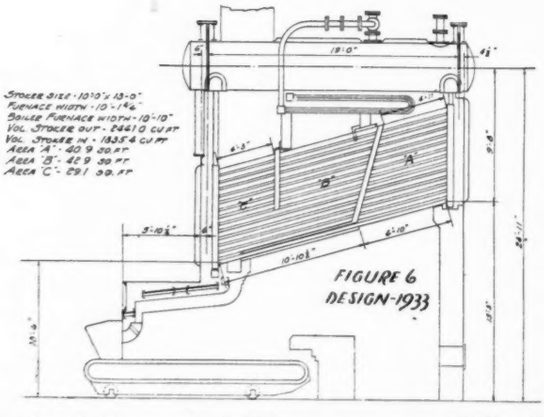
Location	Boone	Boone	Perry	Perry	Perry	Perry	Perry
Rated hp	410	410	500	500	500	500	500
Active grate area	94.5	94.5	126	126	126	126	126
Baffling	Original	Revised	Revised	Revised	Revised	Revised	Revised
Furnace volume	246	386	1,835	1,835	1,835	1,835	1,835
Rating in %	156	180	105	145	158	200	129
Draft over fire	.21	.205	.22	.25	.191	.241	.20
Draft loss through 1st pass	.01	.08	.18	.166	.228	.250	.17
Draft loss through superheater	.22	.12	.061	.05	.060	.065	.05
Draft loss through 2nd pass	.19	.165	.246	.21	.234	.240	.21
Draft loss through 3rd pass	.10	.18	.18	.156	.186	.18	.18
Draft loss differential	.53	.54	.685	.60	.725	.754	.62
Draft stack	.78	.74	.912	.85	.924	1.00	.85
Stack temperature	491	528	486	492	523	553	490
Steam temperature total	491	483	538	549	551	559	540
Steam pressure	170	182	223	227	209	217	225
Steam superheat F	116	103	142	152	160	166	143
Feedwater temp F	212	211	210	210	210	210	210
Length test in hours	3	3	3	3	3	3	24
Pounds of coal burned	13,110	12,670	8400	9700	10,900	14,600	69,000
Pounds of water evap.	78,000	80,000	47,674	63,100	70,952	89,776	463,360
Avg. hourly evap. lb	26,000	26,800	15,891	21,720	23,651	29,925	19,300
Lb of coal burned per sq ft per hr	46.2	44.7	22.2	25.7	28.8	38.6	22.8
Water evap. per lb of coal	5.95	6.3	5.78	6.73	6.5	6.15	6.72
Kind of coal	Scandia	Scandia	Scandia	Scandia	Scandia	Scandia	Scandia
Btu of coal as fired	8617	8462	9190	9190	9018	9018	9219
Total heat of steam	1284	1286	1286	1290	1292	1297	1285
Boiler and furnace eff.	74.2	80.0	70.0	81.3	80.5	76.2	80.4











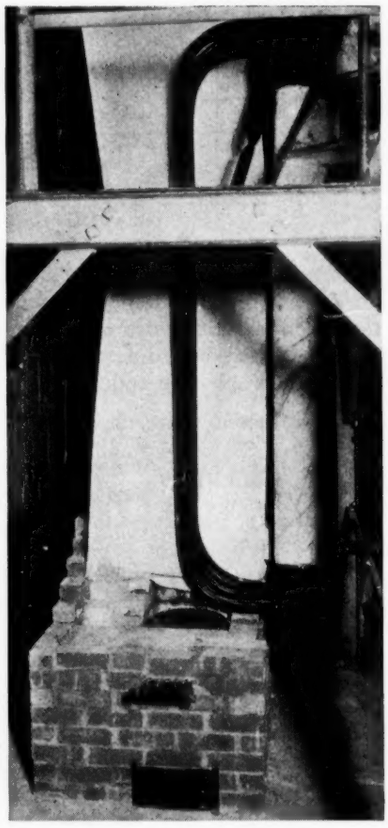
Figs. 2 to 6—Progressive baffling arrangements 1912 to 1933

sults are as shown in Table V. From an examination of this table it will be noted that the data, due to the very short period of testing, is rather inconsistent. We have never given much weight to short tests—twenty-four hours is short enough—but our judgment of performance is really based on month to month performance.

A 1500-Pound Boiler for Experimental Process Work

A most interesting application of high pressure steam to process, where a sustained high temperature is essential, is to be found in a small 1500-lb pressure boiler that a large manufacturer in one of the process industries is now installing for experimental purposes.

The idea underlying the use of saturated steam at this pressure was its ability to supply heat at a constant delivered temperature of 575 F to the particular process. Obviously, superheat and a lower steam pressure would not have provided a sustained temperature during process, for as heat was given up the superheat would drop. If the experiments with steam from this miniature boiler work out satisfactorily it is contemplated to install a full size unit to supply steam at this pressure for regular production. There are undoubtedly processes in other industries wherein this principle could be applied to advantage.



1500-lb pressure experimental boiler before bricking in furnace

The experimental unit, which was built by Combustion Engineering Company, consists of a single row of tubes entering top and bottom headers which are connected by a single downcomer. The tubes are 1 in. outside diameter and the headers of forged steel, rectangular in section, placed 7 ft 2 in. between centers. The total heating surface is slightly over 9 sq ft and the furnace will be a simple refractory lined chamber. The photograph shows the boiler before being bricked in. It is gas fired, of the single-pass type and hand controlled.

Measurement of Nozzle Efficiency

By WINSTON R. NEW

Westinghouse Electric & Mfg. Company

Two methods of determining nozzle efficiency, namely, the reaction balance and the impact traverse, are described and their relative merits discussed. While the reaction balance gives an integrated result for the entire passage the impact method shows what element of the passage is contributing the most harmful effects. Developed formulas are given.

THE most fundamental coefficient with which one must deal in making steam-turbine calculations is nozzle efficiency. This statement is true whether impulse or reaction turbines are considered since the reaction blade passage is merely a non-expanding nozzle. The writer shall define nozzle efficiency as the ratio of the actual kinetic energy of the escaping jet to the energy liberated by an isentropic expansion of the vapor or gas from the initial temperature and pressure to the final pressure, unit masses of the fluid being considered.

There are two fundamentally different and equally effective ways of determining this important coefficient. One method involves the weighing of the force of the jet, and the other method, an impact traverse of the exit plane of the nozzle. The first of these will be referred to as the reaction balance and the second as the impact traverse. The Westinghouse Company uses both methods in the study of steam-turbine nozzles. Elaborate apparatus has recently been installed in its hydrodynamics laboratory at South Philadelphia, and an extensive program of nozzle and blade investigation is now in progress. It is the purpose of this paper to discuss very briefly the measurement of nozzle efficiency by means of this new equipment.

Nozzle Efficiency by Reaction Balance

Suppose a simple de Laval nozzle be mounted at the end of a pipe supported by hollow trunnions as shown in Fig. 1. The pipe swings freely about the center O and there is no friction. A measured mass M of fluid per second is supplied to the nozzle. The reaction force F is determined by adding weights W. Then from the impulse equation can be calculated the actual mean velocity of escape, $\frac{F}{M} = C_2$. Pressure and temperature are measured ahead of the nozzle and the pressure at the nozzle mouth is atmospheric.

The Saint Venant equation for compressible fluids is

$$\frac{C_0^2 - C_1^2}{2g} = \left(\frac{K}{K-1}\right) RT_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{K-1}{K}}\right]$$

where C_0 is the velocity at the end of an isentropic expansion

C_1 is the approach velocity

T_1 is the initial absolute temperature

p_1 is the initial absolute pressure

p_2 is the final absolute pressure

K is the ratio of specific heats, $\frac{C_p}{C_v}$

R is the gas constant.

This equation is solved for C_0. Then the ratio $\frac{C_2}{C_0}$ is the velocity coefficient and the square of this quantity is the energy coefficient or nozzle efficiency.

This apparatus, used by Christlein in his nozzle tests, illustrates the principle, but there are two very serious defects both of which are eliminated in the new Westinghouse tester. The trunnion joints cannot be made both frictionless and gas tight. The line of action of the force F is unknown if a nozzle with beveled mouth is used, because of the jet deflection. In the actual turbine, a beveled mouth nozzle must be used. It is, therefore, imperative that the tests be made with such nozzles.

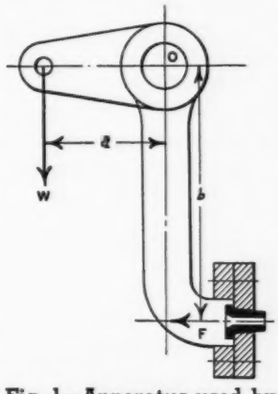


Fig 1—Apparatus used by Christlein for nozzle testing

The reaction balance used in the present studies is shown in Fig 2. The design is believed to be unique in two respects. First, two components of force at right angles are measured, thus permitting the establishment of the reaction in magnitude and direction. Second, the passage supporting the test nozzle floats freely, without mechanical contact, in balanced labyrinths, thus preventing the escape of the metered test fluid without imposing any unknown friction forces.

Referring to Fig 2, the nozzle N is bolted to a passage P at the end of a frame F_1 supported by ball bearings aa. This entire frame is carried on a parallelogram weighing system F_2. A measured mass flow of air enters through the glands LL and is discharged through the nozzle. The mass flow is determined by means of an I.S.A. nozzle* in the inlet piping. The vertical reaction force f_v is measured on the balance in frame F_1 and the horizontal reaction force f_h on the balance in frame F_2.

*The International Standards Association (I.S.A.) nozzle developed in Germany by the V.D.I. and formerly known as the V.D.I. nozzle is a shaped passage whose flow characteristic over a wide operating range is known very accurately. When this type of nozzle is installed and operated under conditions stipulated by the V.D.I., it needs no calibration and gives an indication of the mass flow accurate to within ± ½%.

The efflux angle and total reaction f may then be calculated:

$$f = \sqrt{f_h^2 + f_v^2} \qquad \alpha = \tan^{-1}\frac{f_v}{f_h}$$

Actual and theoretical velocities determined as heretofore described yield nozzle efficiency.

The construction of the glands *LL* is a salient feature of the design. The labyrinth seals are divided by an annular chamber into which an auxiliary supply of air is bled. The pressure in this chamber is balanced against the pressure of the metered air ahead of the labyrinth using a differential manometer. All gland leakage is thus supplied from the auxiliary source and none of the measured air escapes. This method of avoiding a mechanical contact between the piping and the floating head has been found very satisfactory.

Nozzle Efficiency by Impact Traverse

The impact traverse has long been used in establishing velocity profiles in pipes and ducts. The next step to its use in the study of flow through nozzles and blades is one of refinement in method only. Velocity profiles of the nozzle are taken. An integration of these determines the mean velocity. This quantity together with the theoretical velocity fixes the nozzle efficiency.

In Fig 3 is shown a non-expanding nozzle passage formed by two reaction blades. The pitch is shown greatly magnified to facilitate the explanation. An impact tube is arranged so that it may be moved parallel to the exit plane of the nozzle and normal to the nozzle edges. The tube is connected to a pressure recorder in such a way that a continuous line record of the difference between the impact pressure and the pressure in the inlet tank is obtained. Such a record will be somewhat as shown in Fig 3. The impact pressure falls sharply in the wake of the nozzle walls and nearly attains the theoretical value over the central portion of the jet.

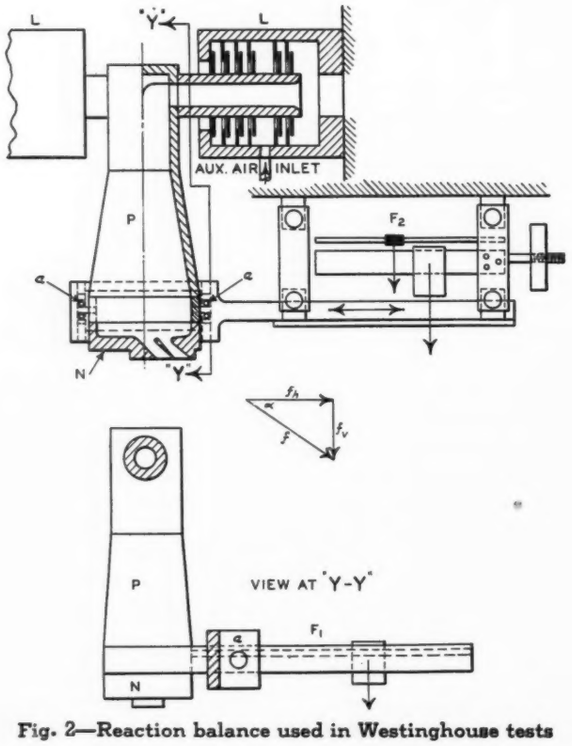


Fig. 2—Reaction balance used in Westinghouse tests

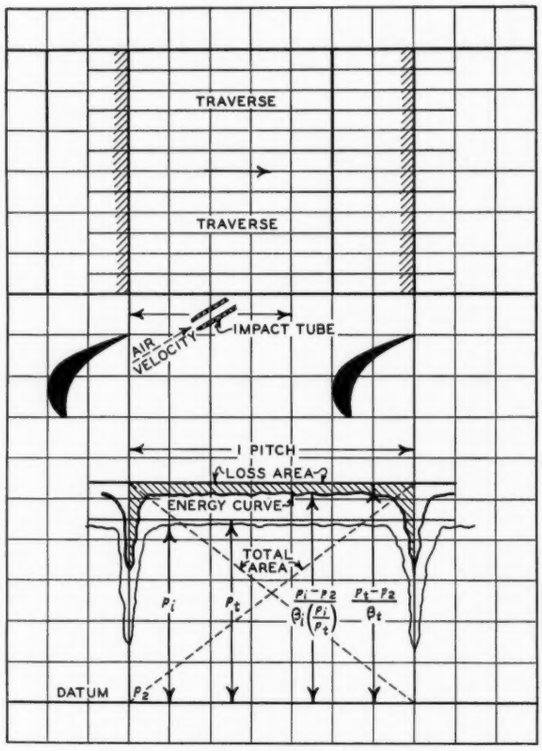


Fig. 3—Efficiency by impact traverse

Since we are dealing with a compressible fluid moving at a velocity in the vicinity of the acoustic velocity we cannot use a record of pressure differences directly as a measure of energy conversion. We must proceed as follows: Rewrite the Saint Venant equation in the form

$$\frac{\gamma_1 C_2^2}{2g}\beta = p_1 - p_2$$

where

$$\beta = \frac{\left(1 - \frac{p_2}{p_1}\right)(K-1)}{K\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{K-1}{K}}\right]}$$

is the compressibility factor, which may be calculated for assumed values of the ratio $\frac{p_2}{p_1}$.

Then $\frac{C_2^2}{2g} = \frac{p_1 - p_2}{\beta\gamma_1}$ where p_2 is atmospheric pressure.

If p_1 and γ_1 are used as the initial pressure and density of the air supplied, then C_2 is the theoretical velocity. Below the acoustic velocity there is no shock loss and it may be assumed that compression in the impact tube is adiabatic. If then we use p_1 as the observed impact pressure and γ_1 as the density in the impact tube where the fluid is brought to rest the value of C_2 will be the actual velocity at the point in question. In this case γ_1 is a variable. Since the fluid starts initially from rest and comes finally to rest at the muzzle of the impact tube without having done external work, the operation is a throttling one, and the final temperatures from point to point are the same and equal to the initial temperature. We may then substitute for γ_1 its equal,

$\frac{p_1}{RT_1}$, where p_1 is now the impact pressure. The expression for efficiency then becomes

$$\frac{C_2^2}{C_0^2} = \frac{p_i - p_2}{\beta_i \frac{p_i}{RT_1}} \div \frac{p_i - p_2}{\beta_i \frac{p_t}{RT_1}} = \frac{p_i - p_2}{\beta_i \left(\frac{p_i}{p_t}\right)} \div \frac{p_t - p_2}{\beta_t}$$

Where p_i is the absolute impact pressure,
p_2 is atmospheric pressure,
p_t is the absolute pressure in the supply tank

The application of this method is quite simple. It is necessary only to divide the ordinates of the pressure records by the proper values of β and the ratio of impact to initial pressure to secure the actual and theoretical energy curves. The area of the loss diagram may be planimetered, the scale of ordinates being linear. Efficiency of the traverse is then $1 - \frac{\text{loss area}}{\text{total area}}$. The process is repeated for a number of narrow strips across the nozzle. Then

Nozzle efficiency = $\sum$ Eff'y of strip $\times$ Area $\div \sum$ Area

In comparative tests, when seeking small differences in efficiency it is a great advantage to be able to measure losses directly. If, for instance, we measure losses with an accuracy of 2 per cent on a passage with an efficiency of 90 per cent, the efficiency measurement is accurate to about $^1/_4$ of 1 per cent, a degree of precision not readily attainable otherwise. This is the greatest single point in favor of the impact method of testing. The reaction balance gives an integrated result for the entire passage. The information is obtained quickly, but is rather limited. The impact test is particularly valuable in determining just what element of a shaped passage is contributing the most harmful effects such as for example, steeply tapered walls and short radii of curvature. Eddies and secondary flow may be analyzed readily.

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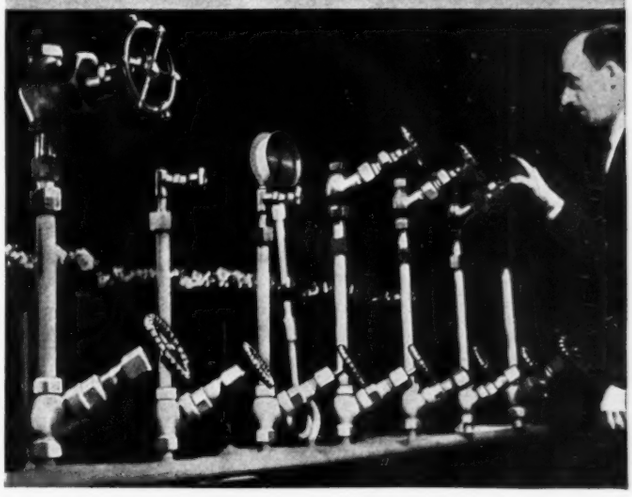
NOT ONE IN 25 ——
—— NOT ONE IN 5
BUT EVERY ONE



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VALVES

REVIEW OF NEW BOOKS

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Elements of Heat-Power Engineering

Parts II and III (Third Edition)

By W. N. Barnard, F. O. Ellenwood and C. F. Hirshfeld

THIS work is in three volumes. Part I covering "Thermodynamics" was completely rewritten in 1926. Parts II and III, covering, respectively, "Steam Generating Apparatus and Prime Movers, Fuels, Combustion and Heat Transmission" and "Auxiliary Equipment, Plant Ensemble, Air Conditioning, and Refrigeration," were completely rewritten in 1933 to form the third edition. In doing this the books have been brought in line with current practice and offer an excellent combination of underlying theory and its reduction to practice. The descriptive range and performance of equipment of different types and makes are unusually complete, typical installations involving a variety of arrangements are included, and the economics of power plant design are discussed.

Two of the authors are well known in the field of engineering instruction as professors of Heat-Power Engineering at Cornell University, and the third is a recognized authority in the field of research and central station power.

Part II contains 871 pages, 6 × 9, 828 illustrations; Part III, 415 pages, 6 × 9, 209 illustrations, and numerous charts and tables. Price: Part II, $5.50; Part III, $4.50.

A.S.T.M. Proceedings—1933

Part I of the 1933 *Proceedings* contains the annual reports of 40 A.S.T.M. standing committees, and research and sectional committees. Seven extensive technical papers are included with the reports.

Among the subjects discussed in reports of interest to the ferrous metals field are the following: steel; wrought iron; cast iron; heat treatment; effect of temperature; corrosion; zinc coating; magnetic properties; malleable iron castings; iron-chromium-nickel and related alloys.

In the section dealing with non-ferrous metals there are reports on corrosion, with extensive data from the results of atmospheric corrosion tests of metals exposed for one year; electrical-heating and electrical-resistance alloys; and die-cast metals and alloys.

During 1933, 47 specifications and methods of testing covering widely used engineering materials were published for the first time. All of these, together with 37 other tentative standards which were revised during 1933, are given in Part I of the 1933 *Proceedings*.

Part II of the 1933 *Proceedings* contains all of the formal technical papers presented at the 1933 annual meeting, together with the extensive oral and written discussion.

Two of the technical contributions, which are of particular interest, are the lecture on "Crystalline Structure in Relation to Failure of Metals—Especially by Fatigue" and the Symposium on Cast Iron. The former was delivered by Dr. H. J. Gough, England, and is based on his extensive research work of over ten years at the National Physical Laboratory. The Symposium was jointly sponsored by the American Foundrymen's Association and the A.S.T.M., and was prepared by a committee of outstanding authorities on cast iron.

Part I contains 1092 pages and Part II, 804; both parts size 6 × 9. Price: $5.50 per part in heavy paper binding; $6.00, cloth; $7.00, half-leather.

Elementary Steam Power Engineering

(Second Edition)

By Edgar MacNaughton

IN preparing a textbook for engineering students it is essential that theory and fundamentals be combined with practice. In this book the usual arrangement is revised and practical phases of the subjects precede the theoretical. While some of the later developments in power plant equipment and practice have been omitted, enough have been included to give the student a good idea of what goes into a power plant, how it functions and its place in the thermal cycle.

In this, the second edition, much of the previous text has been revised, material has been added and the latest steam tables included.

The book contains 649 pages, $5\frac{1}{2} \times 9$, and is cloth bound. Price $5.00.

Technical Thermodynamics

By Professor Dipl.-Ing. W. Schüle

Translated from German by E. W. Geyer

AN excellent reference book covering the thermal properties of gases, the flow of gases, the first and second laws of thermodynamics, deviations from the ideal gas state, special applications of the laws of gases, steam at normal and at high pressures, and general relations of vapors. Many of the thermodynamic problems are presented from a new angle, and the author has included the results of a considerable amount of recent German experimental work. Those engaged in research should find this book especially helpful.

The book contains 627 pages, $5\frac{1}{2} \times 8\frac{1}{2}$, over 300 illustrations and numerous tables. Price $10.00.

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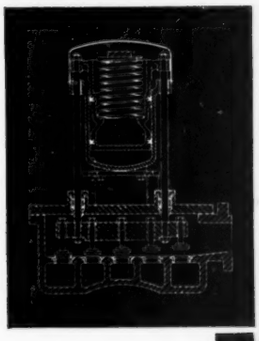

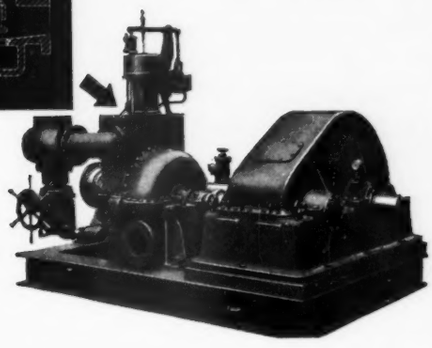

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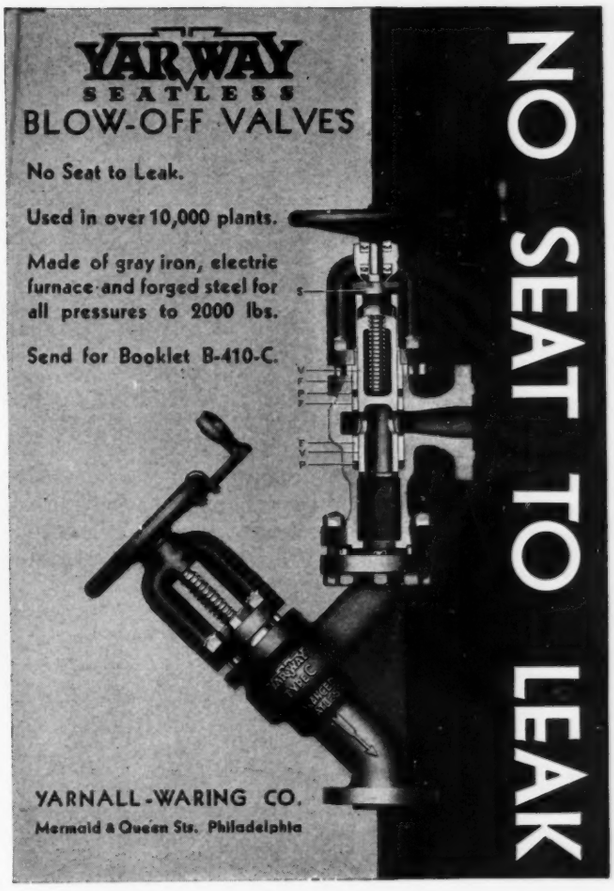
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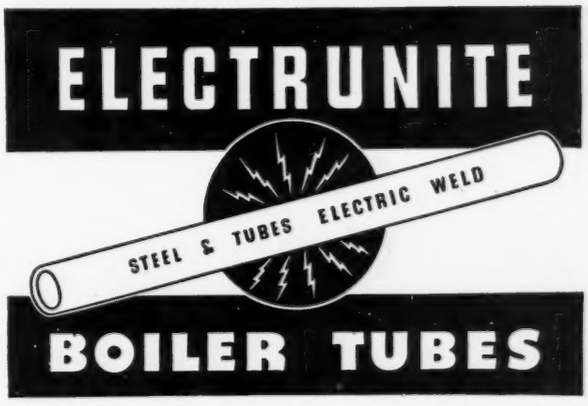

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NEW EQUIPMENT

of interest to steam plant engineers

Blow-off Valve Combination

A blow-off combination in which two valves of entirely different design are

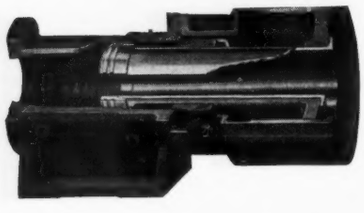
built together is shown in the illustration. This consists of a Hovalco angle valve, at the left, which permits ready accessibility to parts subject to wear, and a Homestead quarter-turn valve, located next to the boiler, and which holds the pressure while parts are being renewed in the angle valve.

For lower pressures the valve bodies are made of semi-steel and for higher pressures cast steel bodies with Monel metal seats, disks and plugs are used.

The combination is manufactured by the Homestead Valve Manufacturing Company of Coraopolis, Pa.

New Expansion Joint

Through an adaptation of the piston ring idea, similar to that used in gas engines, on a new design of slip-type expansion joint of the American District Steam Company, North Tonawanda, N. Y.,

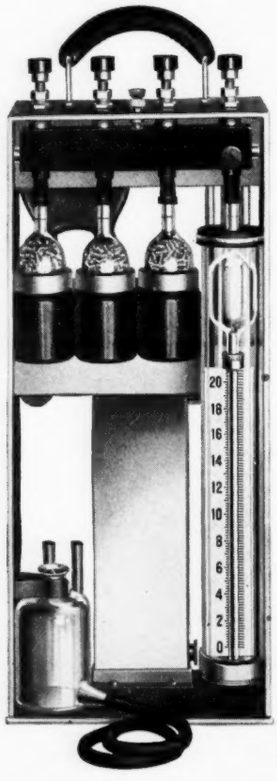


the line pressure in steam or other piping is prevented from reaching the stuffing box during the repacking operation. This permits of completely repacking the joint, if necessary, while the line is operating under full pressure.

Two piston rings are mounted in a piston head securely attached to the inner end of the slip. Back of the piston head in the body is a relief chamber which is vented by piping leading away from the expansion joint. When it is necessary to repack the joint, a valve in the vent piping is opened and the pressure between the piston head and the stuffing box is relieved. Obviously, when the joint is in operation, pressure is built up in the relief chamber. With the vent open this pressure is immediately relieved and such little steam as may pass the piston rings immediately expands, and is discharged through the vent without reaching the stuffing box area. Thus, the joint can be completely repacked without interruption to service.

Portable Gas Analyzer

A new portable gas analyzer of very compact arrangement has recently been designed by Lewis M. Ellison and is being manufactured by the Ellison Draft Gage Company of Chicago. It is furnished



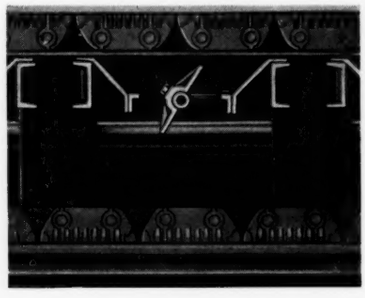
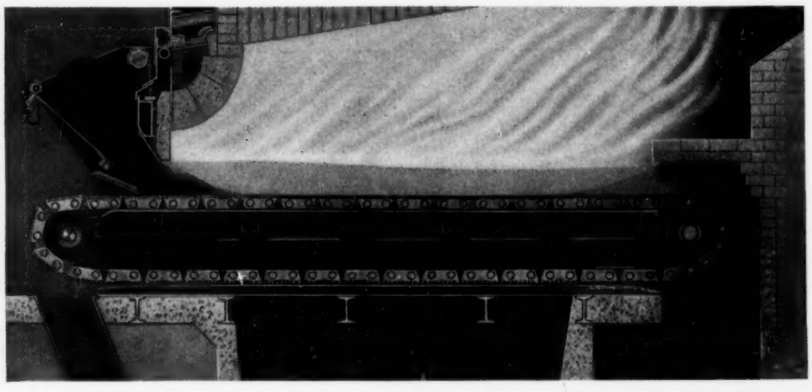
either as a one-chamber analyzer or as a three-chamber analyzer and the case has been so designed as to accommodate a draft gage and a flue gas thermometer.

The carrying case, itself, is of rust resistant furniture steel, dull finish chromium plated, and hard rubber has been employed extensively for various parts of the outfit. The absorption chambers are also filled with curled hard rubber which is a most satisfactory material for surface absorption. All glass parts are annealed. The thermometer holder consists of a packing cup with a 3-in. extension for insertion into the flue gas. It has a mounting stud for which holes are provided in the bottom of the case for carrying one or two thermometers.

The needle valve stem is provided with a swivel joint so that the point does not turn on the seat when closing. In the three-chamber analyzer, the valve stems are of brass with hard rubber points, whereas in the single-chamber analyzer, not using corrosive solution, the valve stem is of nickel.

Green Natural Draft Chain Grate Stoker Redesigned

In order to provide for fixed attachment of controls, a frame construction capable of coping with expansion, and simple conversion to forced draft if desired, Combustion Engineering Company, 200 Madison Avenue, New York, N. Y., has substantially modified the design of its Green Natural Draft Chain Grate Stoker. Instead of a cast-iron frame supported on wheels, as formerly employed, the new design, designated as the "KS," has a stationary structure, fixed into the setting. This consists of transverse channels supported on posts at either side, to which, in turn are fastened angles parallel to the side walls; skids supported on I-beams carry the weight of the chain at the bottom. A pipe, flush with the side wall, and extending nearly the length of the stoker, carries the adjusting rod and takes the tension off the frame. While the take-up in the tension of the chain is at the rear, the actual adjustment is at the front which is always accessible.



With natural draft, zone control may be applied, if desired, by placing louvre dampers, see above, between the channels in one or more compartments. Where conditions make it desirable to change to forced draft, this can readily be accomplished by substituting forced draft links, adding dampers and installing an internal seal and hood at the front.